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# PRACTICAL APPLICATION OF MODERN HYDRODYNAMICS TO MARINE PROPULSION.

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## SECTION I.

### GENERAL THEORY.

1. As indicated by the title, the purpose of this monograph is to apply the principles developed in recent years in the rapid advance of hydrodynamical theory to practical calculations in marine propulsion, and in particular, to the design and analysis of commercial propellers on merchant ships. As modern hydrodynamic theory is an extensive and difficult field, the following presentation is necessarily much compressed. But since an adequate foundation is essential for the development, the fundamental concepts are first presented briefly, together with the simplifying assumptions neces-

sary to make them of practical application in marine engineering. This is supplemented by two practical illustrations, showing the specific application of the theory to actual practice.

2. The hydrodynamics of an ideal fluid dates from the work of Euler and Bernoulli in the middle of the Eighteenth Century, and was developed very completely by Helmholtz, Kirchhoff, Lord Kelvin, and others. This theory dealt with a fluid possessing volume but not viscosity, for the good and sufficient reason that when viscosity was taken into account, the problem became too difficult for solution. Since in an actual fluid such as water or air the action of viscosity is negligible as compared with inertia effects, the assumption of an ideal fluid seemed to be theoretically justified, although it led to the inadmissible conclusion that a solid body moving in such a fluid would experience no resistance to motion. This discrepancy, however, does not invalidate the older theory, since in cases where a body is properly shaped, as in a modern dirigible or blimp, experience has proved that the actual resistance is very slight, and due almost wholly to friction. On the other hand, in cases where form resistance is an essential feature, as for propeller blades or turbine buckets, the older theory is inadequate, and it is necessary to modify it by taking into account the property of viscosity.

3. All actual fluids possess the property of viscosity or internal friction. For the interior of a fluid, however, with the dimensions and velocities ordinarily occurring in practice, the forces due to viscosity are negligible in comparison with those arising from inertia; whereas in the neighborhood of a solid body, forces arising from viscosity are of the same order of magnitude as those due to inertia. Modern hydrodynamics differs from the older theory of an ideal fluid chiefly by taking into account the effect of surface conditions. Surface friction, as ordinarily understood, does not occur, because the layer of fluid in immediate contact with the body is at rest relative to it, and the transition from the velocity of the body to that of the fluid takes place in a thin layer of the fluid, the thickness of which depends on the viscosity. Under certain conditions, this retardation produces a reversal of flow in the boundary layer, leading to the formation of vortices. Whenever this is the case, the theory of non-viscous flow is not applicable without certain modifications.

Although all actual fluids are compressible, the effect of compressibility for the pressure differences ordinarily met in practice is unimportant. As far as practical calculations are concerned, therefore, the fluid may be assumed in general to be incompressible.

4. The method of hydrodynamics consists in expressing in functional relations the velocity of flow and pressure at any arbitrary point in the fluid in terms of the space coordinates of the point, and the time, subject to certain boundary conditions at the limits of the fluid, or surface of a solid immersed in it.

An important idea for the motion of a non-viscous fluid is the line integral of the velocity. This line integral,

$$\int V \cos (V, ds)\, ds$$

taken along the path of a particle from some arbitrary origin to any point P, is called the velocity potential at P. The use of velocity potential simplifies the problem materially, since it enables the velocity to be determined from a single function of the coordinates, and thereby the pressure, $p$, as well, by means of the simple relation known as Bernoulli's equation

$$(1) \qquad p + \frac{\mu}{2g} V^2 = \text{constant},$$

where $\mu$ denotes the specific weight of the fluid, and V is the magnitude of the velocity at the point in question.

In respect to the line integral of the velocity, an important theorem has been established by Lord Kelvin, namely that for a stationary flow, by which is meant one consisting of a continuous succession of particles of an ideal fluid, each particle assuming successively the velocity and pressure of the particle which it replaces, the line integral of the velocity around a closed path does not change with the time. This is equivalent to the theorem that under the given conditions, the line integral of the velocity along any two paths between two points A and B in the fluid, have the same values. From this it follows that any pressure action on the boundaries of the fluid, or arising from bodies immersed in it, can only give rise to potential flow.

The case where the line integral around a closed path in the fluid is different from zero, leads to the famous Helmholtz vortex

theorem. When vortices are present, the line integrals along paths enclosing portions of such space, however limited, in general have a value different from zero. The value of a line integral around a vortex enclosed by the circuit, is called the strength of the vortex; or in general, around a portion of the fluid possessing vortex motion, is called the CIRCULATION. The mathematical and physical consequences of this idea of vortex motion in a viscous fluid, and the resultant idea of circulation, have been developed in various directions by Kutta, Joukowsky, Prandtl, Munk, Betz, Karman, Föttinger, and many other scientists.

5. In many practical applications of hydrodynamics, as in the case of steam and hydraulic turbines, centrifugal pumps, turbo-compressors, and marine and aircraft propellers, the problem in question possesses circular symmetry about a fixed axis. In this event, it is convenient to use cylindrical coordinates. Our starting point, therefore, will be Euler's equations of motion in cylindrical coordinates, $z, r, \varphi$ where $z$ is measured parallel to the fixed axis, $r$ is radial, and $\varphi$ is the angular displacement with respect to a fixed axial plane. Also let $t$ denote time, and $v_z$, $v_r$, $v_\varphi$, the three velocity components in axial, radial and tangential directions, respectively. Then Euler's equations are as follows:

$$(2)\quad \begin{cases} \dfrac{dv_z}{dt} = \dfrac{dv_z}{dt} + v_z\dfrac{dv_z}{dz} + v_r\dfrac{dv_z}{dr} + v_\varphi\dfrac{dv_z}{r\,d\varphi} \\ \dfrac{dv_r}{dt} = \dfrac{dv_r}{dt} + v_z\dfrac{dv_r}{dz} + v_r\dfrac{dv_r}{dr} + v_\varphi\dfrac{dv_r}{r\,d\varphi} \\ \dfrac{dv_\varphi}{dt} = \dfrac{dv_\varphi}{dt} + v_z\dfrac{dv_\varphi}{dz} + v_r\dfrac{dv_\varphi}{dr} + v_\varphi\dfrac{dv_\varphi}{r\,d\varphi} \end{cases}$$

NOTE: In these and all subsequent equations the straight "d" denotes total derivatives and the italic "*d*" denotes partial derivatives.

In addition to the above, we have also the equation of continuity, which expresses the condition for stationary flow, namely, that the same amount flows into a given element of volume per unit of time as flows out from it. This condition may be written in the form

$$(3)\quad \frac{d\,(v_z r)}{dz} + \frac{d\,(v_r r)}{dr} + \frac{d\,(v_\varphi r)}{r\,d\varphi} = 0$$

Introducing the forces, the fundamental dynamical equations are

$$(4)\quad \begin{cases} \frac{\mu}{g}\frac{dv_z}{dt} = P_z - \frac{dp}{dz} \\ \frac{\mu}{g}\left(\frac{dv_r}{dt} - \frac{v_\varphi^2}{r}\right) = P_r - \frac{dp}{dr} \\ \frac{\mu}{g}\left(\frac{dv_\varphi}{dt} + \frac{v_r v_\varphi}{r}\right) = P_\varphi - \frac{dp}{r d\varphi} \end{cases}$$

where $\mu$ denotes the specific weight of the fluid; $p$ is the unit pressure at the point in question; and $P_z$ $P_r$ $P_\varphi$ are the axial, radial and tangential components, respectively, of the body forces acting on unit mass at the given point.

An additional set of relations are those giving the double angular velocity of vortex motion, namely

$$(5)\quad \begin{cases} 2l = \frac{dv_r}{r d\varphi} - \frac{1}{r}\frac{d(v_\varphi r)}{dr} \\ 2m = \frac{dv_\varphi}{dz} - \frac{dv}{r d\varphi} \\ 2n = \frac{dv_z}{dr} - \frac{dv_r}{dz} \end{cases}$$

In particular, the component of the rotation with respect to any axis may be obtained by taking the circulation around an elementary surface of unit area in a plane perpendicular to the axis. In case the fluid rotates like a rigid body, the rotation thus defined comes out as twice the angular velocity of the rigid body.

If we define the strength of a vortex tube as the circulation around its boundary, an important consequence of the definition of rotation is that throughout the entire extent of the vortex tube its strength is constant. Calling the contents of a vortex tube a vortex filament, we have as a consequence of Stokes well known theorem, that the circulation around any closed line is equal to the algebraic sum of the vortex strengths of all the filaments which intersect a surface having the closed line as its boundary. If this closed line is continuously changed so that vortex filaments are thereby cut, then the circulation changes according to the extent of the strength of the vortex filaments cut, which is evidently entirely analogous to certain fundamental theorems of electricity.

In the above equations, the coordinates $z$, $r$, $\varphi$ are referred to fixed axes or planes of reference. The particular problems with which we are concerned, however, involve a rotating unit such as a propeller or turbine runner. In order to reduce the flow picture to that for a planar flow, it is therefore necessary to adopt a system of rotating axes attached to the blade or runner, and regard the flow from the point of view of an observer rotating with these axes.

Denote the corresponding coordinates referred to a set of moving axes attached to the blade or runner by Z, R, $\psi$. Then the relations between the two sets of coordinates are

$$(6) \qquad \begin{cases} Z = z - vt \\ R = r \\ \psi = \varphi - \omega t \end{cases}$$

where $v$ denotes the velocity of flow at a considerable distance in front of the point in question, and $\omega$ is the angular velocity of rotation.

Equations (2) and (3) are of the same form in either set of coordinates. In Equation (4) the body forces in the case of rotating axes consist of gravity and the force of Coriolis. Neglecting gravity as non-essential, we have only the force of Coriolis, with the components

$$(7) \qquad \begin{cases} P_z = 0 \\ P_R = (R\omega^2 + 2R\omega v_\psi)\frac{\mu}{g} \\ P_\psi = -\frac{\mu}{g} 2\omega v_R \end{cases}$$

these components being reckoned positive in the direction of increasing R and increasing $\psi$.

In stationary flow the velocity at any given point does not vary with the time. Consequently the first terms in the right members of Equation (2) are zero, namely,

$$\frac{dZ}{dt} = 0, \frac{dR}{dt} = 0, \frac{d\psi}{dt} = 0.$$

Writing down the dynamical equations analogous to (4) in the moving coordinates, inserting the forces of Coriolis given by (7) and the acceleration components given by equations analogous to (2), we have the following:

$$(8)\quad \begin{cases} \frac{\mu}{g}\left(v_Z\frac{dv_Z}{dZ}+v_R\frac{dv_Z}{dR}+v_\psi\frac{dv_Z}{Rd\psi}\right)=-\frac{dp}{dZ} \\ \frac{\mu}{g}\left(v_Z\frac{dv_R}{dZ}+v_R\frac{dv_R}{dR}+v_\psi\frac{dv_R}{Rd\psi}-\frac{v_\psi^2}{R}\right)=\frac{\mu}{g}\left(R\omega^2+2\omega v_\psi\right)-\frac{dp}{dR} \\ \frac{\mu}{g}\left(v_Z\frac{dv_\psi}{dZ}+v_R\frac{dv_\psi}{dR}+v_\psi\frac{dv_\psi}{Rd\psi}+\frac{v_Rv_\psi}{R}\right)=-\frac{\mu}{g}2\,\omega v_R-\frac{dp}{d\psi} \end{cases}$$

Multiply these relations by dZ, dR, $d\psi$ respectively and add, and simplify the resulting expression by use of (5). Then after reduction, the final expression becomes

$$\frac{\mu}{g}\Big[\Big(2nv_R-2mv_\psi\Big)dZ+\Big\{2\,(1+\omega)\,v_\psi-2\,m\,v_s-R\omega^2-2\,\omega v_\psi\Big\}dR$$

$$(9)\qquad +\Big\{2\,mRv_Z-2\,(1+\omega)\,Rv_R+2\,\omega\,Rv_R\Big\}\,d\psi\Big]$$

$$=-dp-\frac{\mu}{2g}d\,(v^2)$$

where

$$v^2=v_Z^2+v_R^2+v_\psi^2.$$

For the case of cylindrical flow, we have $v_R=0$, $m=0$ and $dR=0$. Consequently the left member vanishes, and the expression reduces to

$$(10)\qquad -dp-\frac{\mu}{2g}d\,(v^2)=0$$

Furthermore, since $v_R=0$, the equation of continuity simplifies into

$$(11)\qquad \frac{dv_Z}{dZ}+\frac{1}{R}\frac{dv_\psi}{d\psi}=0$$

Assuming a potential flow, this relation is satisfied by any holomorphic function of Z and $\psi$, say $\Phi\,(Z,\psi)$. If, then, we put

$$v_Z=\frac{d\Phi}{dZ}\text{ and }v_\psi=\frac{1}{R}\frac{d\Phi}{d\psi}$$

the equation of continuity becomes

$$\frac{d^2\Phi}{dZ^2} + \frac{1}{R^2}\frac{d^2\Phi}{d\psi^2} = 0 \tag{12}$$

The two relations, (10) and (12), show that the dynamics of the relative flow based on the above assumptions is the same as for a two-dimensional potential flow. We have thereby reduced the problem to the much simpler one of stationary two-dimensional flow, which may be regarded as a parallel flow of velocity $V = \sqrt{v^2 + r^2\omega^2}$, which makes an angle $\beta$ with the blade element. *The dynamic action of the propeller then arises from the combination of this parallel flow with circulation around the blade.*

The resultant flow picture arising from this combination is called a Kutta flow.

6. The mathematical treatment of plane potential flow has been worked out more completely than almost any other problem in hydrodynamics. This is due very largely to the use of the powerful method known as CONFORMAL REPRESENTATION, which is widely used in mathematical physics.

The chief features of the method of conformal representation are as follows:

Let $z$ denote the complex variable

$$z = x + iy$$

where $i = \sqrt{-1}$, and let $w = f(z) = u + iv$, where $u$ and $v$ denote holomorphic functions of $x$ and $y$. Then to every point $x + iy$ of the independant variable $z$, will correspond a point $u + iv$ of $w$ or $f(z)$. This point $u + iv$ is called the image of the point $x + iy$. If $w$ is a continuous function of $z$, then every continuous curve in the $z$-plane will have a corresponding image in the $w$-plane, which will also be a continuous curve.

A function is holomorphic in a given region if it is uniform and continuous and possesses a definite continuous derivative at every point in this region.

If the function $w = u + iv$ is holomorphic, then $u$ and $v$ both satisfy Laplace's equation,

$$\frac{d^2V}{dx^2} + \frac{d^2V}{dy^2} = 0 \tag{13}$$

Moreover, the functions $u$ and $v$ have the property that

$$\frac{du}{dx} = \frac{dv}{dy}; \quad \frac{dv}{dx} = -\frac{du}{dy} \tag{14}$$

from which it follows that

$$\frac{du}{dx}\frac{dv}{dx} = -\frac{dv}{dy}\frac{du}{dy} \tag{15}$$

which is the condition that the two families of curves, $u = c_1$, $v = c_2$, where $c_1, c_2$ denote arbitrary constants, intersect everywhere at right angles, or form an orthogonal system. Functions $u$ and $v$ possessing this property are called conjugate functions.

In any conformal system if we construct a set of curves for small equal increments of $u$ and $v$, they divide the plane up into small curvilinear rectangles.

It can also be shown that corresponding curves in the $z$ and $w$ planes intersect each other at the same angle in both corresponding planes. This is called a conformal relation, and is sometimes expressed by saying that conformal representations are similar in their infinitely small parts.

7. In the special case of uniplanar flow, now under consideration as the result of Article 5, the stream lines are all plane curves. In the case of potential flow, every real function $u(x, y)$ and $v(x, y)$ which satisfies the relation

$$u + iv = f(x + iy) \tag{16}$$

where $f$ denotes any holomorphic function, is the potential of such a flow. Moreover, if $u = c_1$ represents the family of equipotential lines, then the curves $v = c_2$ represent the family of stream lines, and conversely. The partial derivatives then represent the velocity components, namely,

$$v_x = \frac{du}{dx} = \frac{dv}{dy}; \quad v_y = \frac{du}{dy} = -\frac{dv}{dx} \tag{17}$$

From what precedes it is evident that the functions $u$, $v$ as well as the complex function $w = u + iv$, satisfy Laplace's equation, (13).

If $z$ represents the point $x$, $y$ of the complex plane, then the velocity of flow at this point is given by

$$\frac{dw}{dz} = \frac{dw}{dx} = \frac{du}{dx} + i\frac{dv}{dx} = v_x - iv_y \tag{18}$$

The function $w$ is the image of the stream lines and equipotential lines of the $z$-plane. If we represent this on another complex plane by means of any holomorphic function

$$Z = f(z),$$

then we obtain a conformal picture of a new potential flow, with velocity $\frac{dw}{dZ}$.

8. The flow around a circular cylinder, or circular cross section for uniplanar flow, has long been known. From this we may deduce the flow around any blade cross section by establishing a conformal relationship between the two by means of the proper coordinating function of the complex variables, as mentioned in Article 7.

If the coordinates in the plane of the circle are $u$ and $v$, and we write $w = u + iv$, then the potential and streamline functions for symmetrical flow around a circular cylinder of diameter $b$ are given by the functional relation

$$\Phi_1 + i\Psi_1 = F\left(w + \frac{b^2}{4w}\right) \tag{19}$$

In particular, the flow function for flow parallel to the $u$ axis may be written

$$\Phi_1 = \frac{c_1}{2}\left(w + \frac{b^2}{4w}\right) \tag{20}$$

where $c_1$ denotes an undetermined parameter.

Interchanging the $u$ and $v$ axes by interchanging $w$ with $iw$, the flow function for flow parallel to the $v$ axis becomes

$$\Phi_2 = \frac{ic_2}{2}\left(w - \frac{b^2}{4w}\right) \tag{21}$$

To these must be added a circulation flow, for which the flow function is

$$\Phi_3 = ic_3 \log w \tag{22}$$

which evidently represents a flow in concentric circles with constant velocity.

The flow function for compound flow with circulation is then represented by the function

$$\Phi = \frac{c_1}{2}\left(w + \frac{b^2}{4w}\right) + \frac{ic_2}{2}\left(w - \frac{b^2}{4w}\right) + ic_3 \log w \tag{23}$$

which, it should be noted, contains three undetermined parameters $c_1, c_2, c_3$.

The flow around any blade profile may be replaced by that around a flat plate giving the same circulation. This gives a very simple type of transformation, namely that established by the coordinating function

$$2z = 2(x + iy) = w + \frac{b^2}{4w} \tag{24}$$

By means of this relation, the circle of diameter $b$ in the $w$ plane becomes transformed into a straight line of length $b$ along the X-axis. Concentric circles, and radii in the w-plane are transformed into confocal systems of ellipses and hyperbolas respectively in the z-plane. (Figure 1.) This gives the flow formation

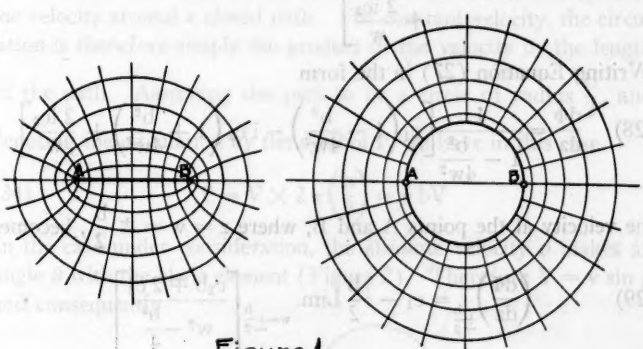


Figure 1.

for a Kutta flow around a segment AB, in which the three undetermined parameters may be so chosen as to determine the velocity and direction of the flow at infinity, and the strength of the circulation.

It is possible by this method of mapping to determine the pressure distribution for any given blade section, and calculated results have been found to agree closely with experimental measurements.

9. The compound flow function given by Equation (23), in connection with Equation (10), makes it possible to investigate the hydrodynamical pressure $p$ in the neighborhood of the blade section.

For this purpose it is necessary first of all to determine the velocity. From Equation (23) we have

$$\frac{d\Phi}{dw} = \frac{c_1}{2}\left(1 - \frac{b^2}{4w^2}\right) - \frac{ic_2}{2}\left(w + \frac{b^2}{4w^2}\right) + \frac{ic_3}{w} \tag{25}$$

and from Equation (24)

$$\frac{dz}{dw} = \frac{1}{2}\left(1 - \frac{b^2}{4w^2}\right) \tag{26}$$

From these we have for the velocity

$$\frac{d\Phi}{dz} = \frac{d\Phi}{dw} \div \frac{dz}{dw} = \frac{1}{1 - \frac{b^2}{4w^2}}\left[\left(c_1 - ic_2\right) - \left(c_1 + ic_2\right)\frac{b^2}{4w^2} + \frac{2\,ic_3}{w}\right] \tag{27}$$

Writing Equation (27) in the form

$$\frac{d\Phi}{dz} = \frac{1}{1 - \frac{b^2}{4w^2}}\left[c_1\left(1 - \frac{b^2}{4w^2}\right) - i\,c_2\left(1 + \frac{b^2}{4w^2}\right) + \frac{2\,ic_3}{w}\right] \tag{28}$$

the velocity at the points A and B, where $z = w = \pm \frac{b}{2}$, becomes

$$\left(\frac{d\Phi}{dz}\right)_{\pm\frac{b}{2}} = c_1 - i\frac{b}{2}\,\text{Lim.}_{w=\pm\frac{b}{2}}\left[\frac{c_2 b \mp 2c_3}{w^2 - \frac{b^2}{4}}\right] \tag{29}$$

Since the $w$ and $z$ axes coincide in direction, from Equation (20) the velocity of flow parallel to the real axis is

$$\frac{d\Phi_1}{dz} = \frac{d\Phi_1}{dw} \div \frac{dz}{dw} = c_1 \tag{30}$$

Similarly from Equation (21), $c_2$ represents the component of the velocity parallel to the axis of imaginaries.

This may also be seen from Equation (27), since at infinity the velocity becomes

$$\left(\frac{d\Phi}{dz}\right)_{w=\infty} = c_1 - ic_2 \tag{31}$$

Let $v$ denote the absolute velocity, and $\beta$ the angle it makes with the blade element. Then

$$c_1 = v\cos\beta \qquad c_2 = v\sin\beta \tag{32}$$

From Equation (29) it is apparent that the velocity increases without limit at both edges A and B provided that the absolute value of the numerator remains finite. In order that the flow should be stable, it is essential that the flow at the trailing edge B, where $z = \frac{b}{2}$, shall take place smoothly. Therefore we choose the circulation parameter $c_3$ so that

$$c_3 = \frac{b}{2}c_2 = \frac{vb}{2}\sin\beta \tag{33}$$

The circulation was defined in Article 4 as the line integral of the velocity around a closed path. For constant velocity, the circulation is therefore simply the product of the velocity by the length of the path. Assuming the path to be a circle of radius $\frac{b}{2}$, and denoting the circulation by the symbol $\Gamma$, we have in this case

$$\Gamma = V \times 2\pi\left(\frac{b}{2}\right) = \pi bV \tag{34}$$

In the case under consideration, the absolute velocity $v$ makes an angle $\beta$ with the blade element (Figure 2). Therefore $V = v\sin\beta$ and consequently

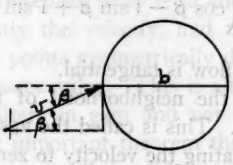


Figure 2.

$$\Gamma = \pi b v \sin \beta = 2 \pi c_3 \tag{35}$$

or

$$c_3 = \frac{\Gamma}{2\pi} \tag{36}$$

The choice of $c_3$ assures smooth flow at the trailing edge B, but at the leading edge A the velocity still remains infinite. *This difficulty is met in practice by rounding off the leading edge.* This result, based on purely theoretical consideration, has been confirmed by experiment in model tanks, where the gain in efficiency due to this source alone has been shown to be in the neighborhood of one per cent.

Substituting the values of $c_1, c_2, c_3$ in Equation (28), the expression for the velocity becomes

$$\frac{d\Phi}{dz} = \frac{v}{1+\frac{b}{2w}}\left[\left(\cos\beta - i\sin\beta\right) + \frac{b}{2w}\left(\cos\beta + i\sin\beta\right)\right] \tag{37}$$

$$= \frac{v}{1+\frac{b}{2w}}\left[\left(\cos\beta - i\sin\beta\right)\left(1+\frac{b}{2w}\right) + \frac{b\, i \sin\beta}{w}\right]$$

Since $\cos\beta - i\sin\beta = e^{-i\beta}$, this may be written in the simpler form

$$\frac{d\Phi}{dz} = ve^{-i\beta} + \frac{v\, i\, b \sin\beta}{w + \frac{b}{2}} \tag{38}$$

At the trailing edge B, where $z = \frac{b}{2}$, the velocity now becomes

$$\left(\frac{d\Phi}{dz}\right)_B = v\left(\cos\beta - i\sin\beta + i\sin\beta\right) = v\cos\beta, \tag{39}$$

and consequently the flow is tangential.

At some point in the neighborhood of the leading edge the velocity becomes zero. This is called the REST POINT, or POINT OF STAGNATION. By equating the velocity to zero, the position of this point is determined by the condition

$$o = e^{-i\beta} + \frac{i\,b \sin \beta}{w + \frac{b}{2}} \tag{40}$$

whence

$$w = -\frac{b}{2} e^{2i\beta} \tag{41}$$

or, since $2z = w + \frac{b^2}{4w}$, we also have

$$z = -\frac{b}{2} \cos 2\beta \tag{42}$$

This indicates that in order to avoid excessive thickness of the leading edge, the angle $\beta$ should be small, say less than 10 degrees. On the other hand, the required thrust fixes a lower limit for $\beta$, so that its value is restricted to rather narrow limits.

11. To investigate the distribution of pressure in the neighborhood of the blade element, consider the pressure at points symmetrically placed with respect to the origin. Since by Equation (10) the hydrodynamical pressure is a function of the velocity, it is sufficient to consider the velocity at such points. For this purpose, separate the velocity into translatory and circulatory parts. Then for translation we have

$$\left(\frac{dw}{dz}\right)_{trans.} = \frac{v}{w^2 - \frac{b^2}{4}} \left( w^2 e^{-i\beta} - \frac{b^2}{4} e^{i\beta} \right) \tag{43}$$

and for circulation

$$\left(\frac{dw}{dz}\right)_{circul.} = \frac{i\Gamma}{\pi} \cdot \frac{w}{w^2 - \frac{b^2}{4}} \tag{44}$$

In Equation (43), if we replace $w$ by $-w$, the expression remains unaltered. Consequently the velocity, and therefore the hydrodynamical pressure, at points symmetrically disposed with respect to the origin, has the same value. This is also true for Equation (44), which merely alters in sign and not in absolute amount. From this follows the important theorem that *neither the translatory motion by itself, nor the circulatory motion by itself, gives rise to any resultant pressure on the blade element.*

This symmetry, however, does not hold for the resultant velocity compounded of both components. The correspondence we have established is such that as a point moves around a circle in the w-plane, the corresponding point moves back and forth in the z-plane along the blade element AB (Figure 3). As $\varphi$ varies from o to $\pi$ we thus traverse one face of the blade element, and as $\varphi$ varies from $\pi$ to $2\pi$, we traverse the opposite face in the reverse direction.

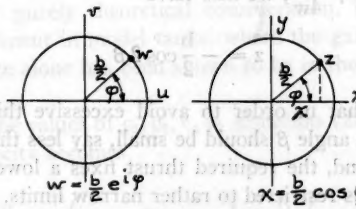


Figure 3.

If we denote the unit pressure on the suction side of the blade element by $p_1$ and on the pressure side by $p_2$, and let $\Delta R_N$ denote the resultant normal pressure on the blade element, then excluding the points A and B from the field of integration by drawing small circles around them of radius $\epsilon$, for an element of length $\Delta r$ we have

$$\Delta R_N = \int_{-\frac{b}{2}+\epsilon}^{+\frac{b}{2}-\epsilon} (p_2 - p_1)\, \Delta r\, dx \tag{45}$$

From Equation (1)

$$p = \text{const.} - \frac{\mu}{2g} v^2 \tag{46}$$

$$= \text{const.} - \frac{\mu}{2g}\left[\left(\frac{dw}{dz}\right)_{trans.} + \left(\frac{dw}{dz}\right)_{circul.}\right]^2$$

$$= \text{const.} - \frac{\mu}{2g}\left[\left(\frac{dw}{dz}\right)^2_{trans.} + \left(\frac{dw}{dz}\right)^2_{circul.} + 2\left(\frac{dw}{dz}\right)_{trans.} \times \left(\frac{dw}{dz}\right)_{circul.}\right]$$

As shown above, the first two (or squared) terms cancel on integration. To evaluate the remaining product term, substitute $w = \frac{b}{2} e^{i\varphi}$ in each factor. Then from Equation (43) we have

$$\left(\frac{dw}{dz}\right)_{trans.} = \frac{v}{e^{i\varphi} - e^{-i\varphi}} \left( e^{i\varphi} e^{-i\beta} - e^{-i\varphi} e^{i\beta} \right) \tag{47}$$

Since $e^{i\varphi} - e^{-i\varphi} = 2\,i \sin \varphi$, and $e^{i\varphi} e^{-i\beta} - e^{-i\varphi} e^{i\beta} = 2\,i \sin (\varphi - \beta)$ this simplifies into

$$\left(\frac{dw}{dz}\right)_{trans.} = \frac{v \sin (\varphi - \beta)}{\sin \varphi} \tag{48}$$

Similarly

$$\left(\frac{dw}{dz}\right)_{circul.} = \frac{\Gamma}{\pi\, b \sin \varphi} \tag{49}$$

Therefore if we exclude the special points A and B from the field of integration as in the case of Equation (45), we have

$$\Delta R_N = \Delta r \int_{-\frac{b}{2}+\epsilon}^{\frac{b}{2}-\epsilon} (p_2 - p_1)\, dx \tag{50}$$

Since $x = \frac{b}{2} \cos \varphi$, we have $dx = -\frac{b}{2} \sin \varphi\, d\varphi$ and consequently by reason of Equations (48) and (49), in connection with (46), we find

$$\left\{ \begin{aligned} \Delta R_N &= \frac{v\,\mu\,\Gamma\,\Delta r}{2\,\pi g} \operatorname{Lim}_{\epsilon \doteq 0} \left[ \int_{\epsilon}^{\pi-\epsilon} \frac{\sin (\varphi - \beta)}{\sin \varphi}\, d\varphi \right. \\ &\left. + \int_{\pi+\epsilon}^{2\pi-\epsilon} \frac{\sin (\varphi - \beta)}{\sin \varphi}\, d\varphi \right] \\ &= \frac{v\,\mu\,\Gamma\,\Delta r}{2\,\pi g} \left( 2\,\pi \cos \beta \right) = \frac{v\,\mu\,\Gamma\,\Delta r \cos \beta}{g} \end{aligned} \right. \tag{51}$$

or, since

$$= \pi\, b v \sin \beta \tag{52}$$

this may also be written as

$$\Delta R_N = \frac{\pi b \mu v^2 \Delta r}{g} \sin \beta \cos \beta \tag{53}$$

which gives the required elementary normal pressure.

12. A much simpler method of obtaining this result is by means of the Kutta-Joukowsky theorem. This theorem states that the lift $\Delta L$ acting on an element of length $\Delta r$ of blade (or wing) is given by the simple formula

$$\Delta L = \frac{\mu}{g} \Gamma v \Delta r \tag{54}$$

where $\Gamma$ denotes the circulation, and the direction of the lift is perpendicular to the direction of the velocity.

As indicated in Figure 4, the normal component of this resultant is then

$$\Delta R_N = \Delta L \cos \beta \Delta r = \frac{\mu}{g} \Gamma v \cos \beta \Delta r$$

which agrees with Equation (51).

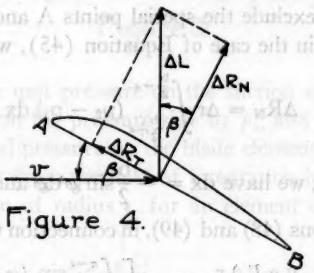


Figure 4.

For the tangential component $\Delta R_T$ we have

$$\Delta R_T = \Delta L \sin \beta \Delta r = \frac{\mu}{g} \Gamma v \sin \beta \Delta r$$

These results are of special importance in what follows.

## SECTION II.

### APPLICATION TO PROPELLER DESIGN.

13. In what precedes we have considered the general problem of flow, and by means of a simple correspondence based on the well known method of conformal representation in the plane of the complex variable, have reduced the somewhat involved case of compound flow to a simple planar flow involving circulation. This

represents the theoretical groundwork of the subject, as established by the work of various well-known scientists.

In applying these results to marine propellers, we will take as our starting point the method of analysis indicated by Grammel. This consists in deriving the indefinite integrals (73) and (74) as given in Article 13, and the assumption of a definite expression for the circulation, in order to make the integrands known functions of $r$, as given in Article (14). The rest of this monograph, comprising the evaluation of Grammel's integrals, and the derivation of complete functional relations for thrust and torque in terms of non-dimensional combinations of the various quantities involved, as well as the practical application of these relations to commercial propellers, is original with the author and heretofore unpublished.

Assume that on account of friction, the normal and tangential components attain only a certain fraction of their theoretical value, say $K_1 \Delta R_N$ and $K_2 \Delta R_T$ where $K_1, K_2$ denote proper fractions. Then the angle $\delta$ which the new resultant makes with the normal is given by the relation

$$\tan \delta = \frac{K_2}{K_1} \tan \beta \tag{56}$$

as indicated in Figure 5, and the magnitude of this new resultant is

$$\Delta R = \frac{K_1 \cos \beta}{\cos \delta} \Delta L \tag{57}$$

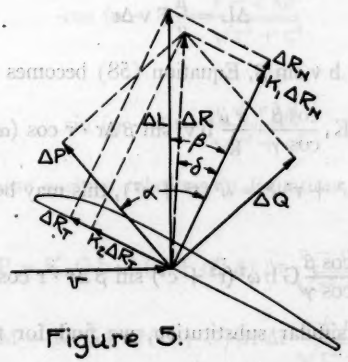


Figure 5.

To determine the thrust and torque it is necessary to resolve $\Delta R$ into components P and Q, respectively parallel and perpendicular to the axis of the propeller shaft. Referring to Figure 5, we have

$$\Delta P = \Delta R \cos(\alpha - \delta) = \frac{K_1 \cos\beta}{\cos\delta} \Delta L \times \Delta r \sin(\alpha - \delta) \tag{58}$$

Similarly, if $r$ denotes the radius of the blade element, measured from the shaft axis, the element of torque is

$$\Delta M = r\,\Delta Q = r\,\Delta R \sin(\alpha - \delta) = K_1 \frac{\cos\beta}{\cos\delta} \Delta L \cdot \Delta r \sin(\alpha - \delta) \tag{59}$$

Let $\omega$ denote the angular velocity of rotation of the blade about the shaft axis. Then if V denotes the translational velocity of the propeller, or its velocity of advance, we have

$$v^2 = V^2 + r^2\omega^2 \tag{60}$$

Now for simplicity, let

$$c = \frac{V}{\omega} \tag{61}$$

where we may call $c$ the coefficient of advance; and also let

$$G = \frac{\pi\mu}{g} \tag{62}$$

From Equation (54) we also have

$$\Delta L = \frac{\mu}{g} \Gamma v \Delta r$$

and since $\Gamma = \pi\ b\ v \sin\beta$, Equation (58) becomes

$$\Delta P = K_1 \frac{\cos\beta}{\cos\gamma} \cdot \frac{\pi\mu}{g} b v^2 \sin\beta \Delta r \cdot r \cos(\alpha - \gamma) \tag{63}$$

or, since $v^2 = c^2\omega^2 + r^2\omega^2 = \omega^2(r^2 + c^2)$, this may be written in the form

$$\Delta P = K_1 \frac{\cos\beta}{\cos\gamma} G b \omega^2 (r^2 + c^2) \sin\beta \Delta r \cdot r \cos(\alpha - \gamma) \tag{64}$$

By means of a similar substitution, we find for the element of torque

$$(65)\quad \Delta M = K_1 \frac{\cos\beta}{\cos\gamma} G\, b\, \omega^2 (r^2 + c^2) \sin\beta\, \Delta r \cdot r \sin(\alpha - \gamma)$$

In all cases arising in practice, the angle $\beta$ is a small angle independent of the radius, lying between the limits, say, of 2 degrees and 8 degrees. We will therefore incur only a negligible error if we replace $\frac{\sin\beta\cos\beta}{\cos\delta}$ by $\beta$. We may also introduce an auxiliary parameter $\lambda$ such that

$$(66)\quad \beta - \delta = \lambda\beta$$

Then $\alpha - \delta = (\alpha - \beta) + (\beta - \delta) = (\alpha - \beta) + \lambda\beta$ and consequently

$$(67)\quad \cos(\alpha - \delta) = \cos(\alpha - \beta)\cos\lambda\beta - \sin(\alpha - \beta)\sin\lambda\beta,$$

or, since

$$\cos(\alpha - \beta) = \frac{r\omega}{V} = \frac{r}{\sqrt{r^2 + c^2}} \text{ and } \sin(\alpha - \beta) = \frac{V}{v} = \frac{c}{\sqrt{r^2 + c^2}},$$

this may be written

$$(68)\quad \cos(\alpha - \delta) = \frac{r\cos\lambda\beta}{\sqrt{r^2 + c^2}} - \frac{c\sin\lambda\beta}{\sqrt{r^2 + c^2}}$$

Since $\lambda\beta$ is small, we have approximately $\cos\lambda\beta = 1$ and $\sin\lambda\beta = \lambda\beta$, and therefore finally

$$(69)\quad \cos(\alpha - \delta) = \frac{r - c\lambda\beta}{\sqrt{r^2 + c^2}}$$

Similarly

$$(70)\quad \sin(\alpha - \delta) = \frac{c + r\lambda\beta}{\sqrt{r^2 + c^2}}$$

Substituting these various relations in Equations (64) and (65), they become

$$(71)\quad \Delta P = K_1 G\, b\, \omega^2 (r^2 + c^2)\, \beta\, r\, \Delta r \frac{r - c\lambda\beta}{\sqrt{r^2 + c^2}}$$

$$(72)\quad \Delta M = K_1 G\, b\, \omega^2 (r^2 + c^2)\, \beta\, r\, \Delta r \frac{c + r\lambda\beta}{\sqrt{r^2 + c^2}}$$

Consequently the required expressions for total thrust and torque are obtained by integrating these relations over the entire blade, from the hub radius $r_0$ to the tip radius $r_1$, namely

$$(73) \qquad P = K_1 G \omega^2 \beta \int_{r_0}^{r_1} b \sqrt{r^2 + c^2} (r - c\lambda\beta)\, dr,$$

$$(74) \qquad M = K_1 G \omega^2 \beta \int_{r_0}^{r_1} b \sqrt{r^2 + c^2} (r\lambda\beta + c)\, dr,$$

which are accordingly the Grammel integrals mentioned above.

14. In discussing these expressions for thrust and torque, Grammel made no attempt to carry out the integration, but contented himself with seeking the upper and lower limits of the indefinite integrals as dependant on the number of blades. The result of his investigations showed that when the number of blades was small, as in the case of marine or aircraft propellers, the upper and lower limits of the integrals lay close together, so that the values of thrust and torque given by Equations (73) and (74) were sufficiently accurate for all practical purposes.

A more recent method of attacking the question of blade interference, consists in regarding the circulation around a blade as a vortex filament of equivalent strength, and investigating the flow around a single blade in the presence of stationary vortices representing the influence of the other blades. The result shows that when the number of blades is small, as in the case of propellers, the effect of blade interference is to alter the circulation slightly, or what amounts to the same thing, to make a slight change in the angle of incidence.

It is evident therefore that since the expressions for thrust and torque still involve an undetermined parameter $K_1$, which must be found by experiment, these relations may be applied directly to propeller design without any loss in generality. The noteworthy fact is that these relations are rational, and based on sound mathematical and physical principles. Consequently they furnish a true insight into the major facts of propulsion and clear up many facts which have always been obscured by the limitations of empirical methods.

In Equations (73) and (74), the blade width $b$ is a function of the radius $r$. Consequently the form of this function must be known before the integration can be performed.

The most rational method is to assume the law of variation of the circulation along the blade, and thereby determine $b$ in terms of $r$. From Equation (35) the circulation is $\Gamma = \pi\, b\, v \sin\beta$, or, since $v = \omega\sqrt{r^2 + c^2}$, this becomes

$$\Gamma = \pi\, b\, \omega \sin\beta \sqrt{r^2 + c^2} \tag{75}$$

Since the thrust is a function of the circulation, and the thrust becomes zero at the tips of the blades, it is desirable to assume an expression for the circulation which shall also approach zero as $r$ approaches the tip radius $r_1$. We may therefore assume as the law by which the circulation varies radially along the blade, the function

$$\Gamma = A\, r^m (r_1 - r)^n \tag{76}$$

where A is an undetermined parameter, and $m$ and $n$ denote positive numbers, either fractional or integral. The condition that this assumption shall be consistent with the value of the circulation as determined by Equation (75) is evidently that

$$b\sqrt{r^2 + c^2} = \frac{A\, r^m (r_1 - r)^n}{\pi\, \omega \sin\beta} \tag{77}$$

To simplify this condition, introduce a new parameter B such that

$$B = \frac{A}{\pi\, \omega \sin\beta} \tag{78}$$

In this case, the relation given by Equation (77) becomes

$$b\sqrt{r^2 + c^2} = B\, r^m (r_1 - r)^n. \tag{79}$$

Introducing this relation in Equations (73) and (74), the integrands become functions of $r$ alone, and consequently the integration may be performed as soon as proper values are assigned to the exponents $m$ and $n$.

15. The constants $m$ and $n$ determine the form of blade profile. To cover the range of marine propeller design, from narrow tipped to wide tipped blades, the writer has chosen six pairs of values of $m$ and $n$, and carried out the calculations in full for each of the six types of blade profile so determined. The following table gives the values of $m$ and $n$ for these six types of profile.

| Blade Profile | | |
|---|---|---|
| Type | m | n |
| A | 1 | 1/2 |
| B | 1 | 1/4 |
| C | 3/2 | 1/2 |
| D | 3/2 | 1/4 |
| E | 2 | 1/2 |
| F | 2 | 1/4 |

The blade parameter B, given by Equation (78), determines the actual width of blade in any given case. By varying this parameter B in connection with the six types of profile listed above, the entire range of practical design is fully covered.

Type E has been found to be especially suitable for commercial propellers for merchant ships. The integrations will therefore be carried out in full for this particular type. For the other five types, the work will not be given in detail for lack of space, but only the final results tabulated.

16. For Type E of blade profile we have $m = 2$, $n = 1/2$. For these values, Equation (79) becomes

$$b\sqrt{r^2 + c^2} = B r^2 (r_1 - r)^{1/2}. \tag{80}$$

Substituting this relation in Equations (73) and (74), they become

$$\begin{cases} P = K_1 G \omega^2 \beta B n \displaystyle\int_{r_0}^{r_1} r^2 (r_1 - r)^{1/2} (r - c\lambda\beta)\, dr; \\ M = K_1 G \omega^2 \beta B n \displaystyle\int_{r_0}^{r_1} r^2 (r_1 - r)^{1/2} (r\lambda\beta + c)\, dr; \end{cases} \tag{81}$$

where now $n$ denotes the number of blades.

The integrations are somewhat long and tedius, and are performed by successive applications of reduction formulas. Without giving this work in detail, the final results in the present case are as follows:

$$(82)\quad \begin{cases} P = K_1 G n \omega^2 \beta B \left[ \dfrac{r_0^3 (r_1 - r_0)^{3/2}}{4\frac{1}{2}} + \left( \dfrac{2}{3} r_1 - c\lambda\beta \right) \right. \\ \qquad \left. \left\{ \dfrac{2}{105} (8 r_1^2 + 12 r_1 r_0 + 15 r_0^2)(r_1 - r_0)^{3/2} \right\} \right] \\ M = K_1 G n \omega^2 \beta B \left[ \dfrac{\lambda\beta r_0^4 (r_1 - r_0)^{3/2}}{5\frac{1}{2}} + \left( \dfrac{8 r_1 \lambda\beta}{11} + c \right) \right. \\ \qquad \left. \left\{ \dfrac{r_0^3 (r_1 - r_0)^{3/2}}{4\frac{1}{2}} + \dfrac{3 r_1}{4\frac{1}{2}} \left( \dfrac{2}{105} (8 r_1^2 + 12 r_1 r_0 + 15 r_0^2) \right. \right. \right. \\ \qquad \left. \left. \left. (r_1 - r_0)^{3/2} \right) \right\} \right] \end{cases}$$

In marine practice, the hub diameter in the case of a built up propeller with separate blades may be assumed as 25 per cent of the propeller diameter, or $r_0 = .25\, r_1$. Substituting this value for $r_0$, the formulas for thrust and torque for the case of separate blades become

$$(83)\quad \begin{cases} P = K_1 G n \omega^2 \beta B r_1^{7/2} \left[ .100713\, r_1 - .1477\, c\lambda\beta \right]; \\ M = K_1 G n \omega^2 \beta B r_1^{9/2} \left[ .0737137\, \lambda\beta r_1 + .100713\, c \right]. \end{cases}$$

For a solid wheel, we may assume the hub radius as $r_0 = .20\, r_1$, in which case the formulas become

$$(84)\quad \begin{cases} P = K_1 G n \omega^2 \beta B r_1^{7/2} \left[ .101222\, r_1 - .1499\, c\lambda\beta \right]; \\ M = K_1 G n \omega^2 \beta B r_1^{9/2} \left[ .0738242\, \lambda\beta r_1 + .101222\, c \right]. \end{cases}$$

17. In order to make these results of general application, it is essential that they be expressed in terms of non-dimensional com-

binations of the various quantities involved. For this purpose we will make use of the principle of dimensional homogeneity, which states that all the terms of any complete and correct physical equation must have the same dimensions. The chief value of this principle is that it enables us to determine *a priori*, the qualitative relations existing between the physical quantities involved in any actual problem, and thus makes it possible to give a rational form to empirical equations. In the present case it is equally useful in transforming the somewhat cumbersome integrals we have obtained for thrust and torque, into simple non-dimensional formulas.

The basis of this method is a theorem first published by Riabouchinski, and generally referred to as the product theorem. This theorem may be stated as follows:

Suppose we know that any physical problem under consideration involves a certain number, $n$, of dimensional quantities, which we will denote by $Q_1\, Q_2 \ldots\ldots Q_n$. Then the physical law, or relation, existing between these quantities, which we are seeking to determine, will be represented by some function of these quantities, say

$$f(Q_1 Q_2 \cdot \cdot \cdot Q_n) = 0. \tag{85}$$

Then, if $k$ denotes the number of independent units necessary to specify the Q's, the general theorem states that any such functional relation may be reduced to an equivalent relation involving $n$-$k$ dimensionless products of the Q's, say

$$F(\Pi_1 \Pi_2 \cdot \cdot \cdot \Pi_{n-k}) = 0 \tag{86}$$

where each of the $\Pi$'s represents a dimensionless product of the Q's of the form

$$\Pi = Q_1^x Q_2^y \cdot \cdot \cdot Q_n^s \tag{87}$$

To apply this theorem to the problem in hand, consider first the propeller thrust P. This we know depends primarily on the diameter D, the speed of advance V, speed of revolution N, and density $\sigma$ of the fluid in which it works. For the density $\sigma$ we have $\sigma = \frac{\mu}{g}$, where $\mu$ denotes the specific weight of the fluid, and $g$ is the acceleration of gravity.

The three fundamental units in terms of which we choose to measure these five quantities are length, time and mass, which we will denote by the symbols L, T and M. Then the dimensions of the quantities involved are as follows:

| Quantity | Symbol | Dimensions |
|---|---|---|
| Diameter | D | L |
| Speed | V | $L\,T^{-1}$ |
| Revolutions | N | $T^{-1}$ |
| Thrust | P | $M\,L\,T^{-2}$ |
| Torque | M | $M\,L^{2}\,T^{-2}$ |
| Density | $\sigma$ | $M\,L^{-3}$ |

We thus have $n = 5$ dimensional quantities measured in terms of $k = 3$ fundamental units. The theorem states then that any functional relation in the quantities involved, say

$$f(D, V, N, P, \sigma) = o \tag{88}$$

may be reduced to an equivalent expression involving $n - k = 2$ non-dimensional products, say

$$F(\Pi_1 \Pi_2) = o. \tag{89}$$

To apply this in the present case, select any three of our five dimensional quantities, say D, V and $\sigma$, and combine these with the remaining two, namely P and N, in the form of two products, namely

$$\begin{cases} \Pi_1 = D^x V^y \sigma^z P \\ \Pi_2 = D^u V^v \sigma^w N \end{cases} \tag{90}$$

We have now to determine the exponents so that each product shall be dimensionless. Inserting the dimensions in $\Pi_1$ it takes the dimensional form

$$(L)^x (LT^{-1})^y (ML^{-3})^z (MLT^{-2})$$
$$= L^{x+y-3z+1} M^{z+1} T^{-y-2}$$

Therefore the conditions which must be satisfied if $\Pi_1$ is to be dimensionless, are

$$(91)\qquad \begin{cases} x+y-3z+1=o \\ z+1 \qquad\quad =o \\ -y-2 \qquad\quad =o \end{cases}$$

whence solving algebraically, we find

$$x=-2,\quad y=-2,\quad z=-1$$

Consequently

$$\Pi_1=\frac{P}{D^2V^2\sigma}$$

Proceeding similarly with $\Pi_2$ we find

$$\Pi_2=\frac{DN}{V}$$

The general relation sought must therefore be of the form

$$(92)\qquad F\left(\frac{P}{D^2V^2\sigma},\ \frac{DN}{V}\right)=o$$

or, solving for $\Pi_1$, we may write this in the more convenient form

$$(93)\qquad \frac{P}{D^2V^2\sigma}=\varphi\left(\frac{DN}{V}\right)$$

In the case of the torque, M, we again have five dimensional quantities, M, D, V, N, $\sigma$, expressed in terms of three fundamental units, L, T, M, so that here also $n-k=2$. Assuming for our two products

$$(94)\qquad \begin{cases} \Pi_1=D^xV^y\sigma^s M \\ \Pi_2=D^uV^v\sigma^w N \end{cases}$$

forming the dimensional equations, and solving the resulting conditions in the exponents that the $\Pi$'s shall be dimensionless, we find in the present case

$$(95)\qquad \begin{cases} \Pi_1=\dfrac{M}{D^3V^2\sigma} \\ \Pi_2=\dfrac{DN}{V} \end{cases}$$

Hence our general functional relation becomes

$$F\left(\frac{M}{D^3 V^2 \sigma}, \frac{D N}{V}\right) = o \tag{96}$$

or, solving for $\Pi_1$,

$$\frac{M}{D^3 V^2 \sigma} = \psi\left(\frac{D N}{V}\right). \tag{97}$$

This method does not show that the relation so obtained is complete. In other words, the physical equation sought may involve other non-dimensional quantities which do not appear explicitly. This happens in the present case, as will appear shortly.

18. We will now proceed to transform Equations (83) and (84) into the form indicated by Equations (93) and (97).

In the former, we have $G = \frac{\pi \mu}{g}$ and in the latter $\sigma = \frac{\mu}{g}$. Therefore; since $\pi$ is an abstract number, $\sigma$ is of the same dimensions as G. Consequently we seek to rearrange the terms and factors in Equations (83) and (84) so that their left members shall take the forms $\frac{P}{D^2 V^2 G}$ and $\frac{M}{D^3 V^2 G}$ respectively, and their right members shall be functions of $\frac{D N}{V}$.

Starting with Equations (83) for built-up propellers with blades separate from the hub substitute

$$r_1 = \frac{D}{2}, \omega = \frac{2 \pi N}{60} \text{ and } c = \frac{V}{\omega} = \frac{60 V}{2 \pi N},$$

where N denotes revolutions per minute and all other quantities are expressed in foot and second units. Also denote by S the non-dimensional product $B D^{1/4}$. Then combining numerical factors, Equations (83) finally become

$$\begin{cases} \frac{P}{G V^2 D^2} = K_1 n \beta S \frac{D N}{V}\left(.000048816 \frac{D N}{V} - .0013671 \lambda \beta\right) \\ \frac{M}{G V^2 D^3} = K_2 n \beta S \frac{D N}{V}\left(.000017863 \lambda \beta \frac{D N}{V} + .0004662\right) \end{cases} \tag{98}$$

Proceeding similarly with Equations (84) for solid wheels, they become after reduction

$$(99)\quad \begin{cases} \dfrac{P}{G\,V^2\,D^2} = K_1\, n\, \beta\, S\, \dfrac{D\,N}{V}\left(.00004906\,\dfrac{D\,N}{V} - .001388\,\lambda\beta\right) \\ \dfrac{M}{G\,V^2\,D^3} = K_2\, n\, \beta\, S\, \dfrac{D\,N}{V}\left(.00001789\,\lambda\beta\,\dfrac{D\,N}{V} + .0004684\right) \end{cases}$$

From these relations it is apparent that within the limits customary in practice, the torque is affected but slightly by hub diameter, and the thrust to a somewhat greater extent. The argument for using small hubs is based on decreased appendage resistance and wake formation rather than on increase in blade efficiency.

19. By a similar process of integration and reduction to non-dimensional form, the thrust and torque characteristics have been derived for all six types of blade profile tabulated in Article 15. These differ only in the numerical constants involved, and in what we may call the shape factor, S, which is a non-dimensional quantity of the form $B\,D^k$. The value of $k$ determines the type of blade profile, and the value of B determines the actual blade width for the type in question.

The resulting characteristics for all six types of blade profile are summarized in the following tabular form for convenience.

These formulas differ essentially from so-called propeller characteristics in use heretofore in that they exhibit a complete functional relation between the quantities involved. It has been customary heretofore to form some arbitrary combination of variables involving thrust, and another involving torque, and plot these relations as determined by experiment in terms of slip. The results then appeared in the form of separate curves for each area ratio and each pitch ratio.

In the present case we have established one general formula for thrust and one for torque. The functional form of these relations has been determined in two ways, entirely independent of each other, namely, by direct analysis, and by the theory of dimensions. The results are therefore physically rigorous as to form, and absolutely general in application.

Moreover, the fact that the combinations of variables involved are all non-dimensional, makes it possible to apply the formulas directly either to small scale models or to full scale practice. Furthermore, they are not restricted to one particular class of

vessels as in the case of empirical formulas, but apply equally well to all speeds and dimensions throughout the entire range of design.

It may be noted, however, that these formulas are not intended to apply to aircraft propellers. The latter are not subject to the same mechanical limitations as marine propellers, and therefore in applying the circulation theory to aircraft propellers we proceed somewhat differently from the method used in what precedes, and

## PROPELLER CHARACTERISTICS

THRUST $\frac{P}{GV^2D^2} = K_1 \eta \beta S \frac{DN}{V}\left(a_1 \frac{DN}{V} - a_2 \lambda\beta\right)$

TORQUE $\frac{M}{GV^2D^3} = K_2 \eta \beta S \frac{DN}{V}\left(b_1 \lambda\beta \frac{DN}{V} + b_2\right)$

| TYPE | Thrust Constants | | Torque Constants | | Shape Parameter | Blade Width |
|---|---|---|---|---|---|---|
| | $a_1$ | $a_2$ | $b_1$ | $b_2$ | $S = BD^K$ | $b = \frac{Br^m\left(\frac{D}{2}-r\right)^n}{\sqrt{r^2+c^2}}$ |
| A | .0001453 | .004592 | .00004906 | .001388 | $S = BD^{-\frac{1}{2}}$ | $b = \frac{Br\left(\frac{D}{2}-r\right)^{\frac{1}{2}}}{\sqrt{r^2+c^2}}$ |
| B | .0002493 | .007403 | .00008879 | .002380 | $S = BD^{-\frac{3}{4}}$ | $b = \frac{Br\left(\frac{D}{2}-r\right)^{\frac{1}{4}}}{\sqrt{r^2+c^2}}$ |
| C | .00008347 | .002478 | .00002939 | .0007970 | $S = B$ | $b = \frac{Br^{\frac{3}{2}}\left(\frac{D}{2}-r\right)^{\frac{1}{2}}}{\sqrt{r^2+c^2}}$ |
| D | .0001461 | .004068 | .00005437 | .001395 | $S = BD^{\frac{1}{4}}$ | $b = \frac{Br^{\frac{3}{2}}\left(\frac{D}{2}-r\right)^{\frac{1}{4}}}{\sqrt{r^2+c^2}}$ |
| E | .00004906 | .001388 | .00001789 | .0004684 | $S = BD^{\frac{1}{2}}$ | $b = \frac{Br^2\left(\frac{D}{2}-r\right)^{\frac{1}{2}}}{\sqrt{r^2+c^2}}$ |
| F | .00008878 | .002380 | .00003390 | .0008482 | $S = BD^{\frac{3}{4}}$ | $b = \frac{Br^2\left(\frac{D}{2}-r\right)^{\frac{1}{4}}}{\sqrt{r^2+c^2}}$ |

derive the required characteristics by applying directly the calculus condition for maximum efficiency. For aircraft propellers it is possible to obtain a more favorable aspect ratio than is possible for marine propellers, together with a very much lighter unit load, and smaller angle of incidence, as well as fewer blades and therefore less interference, all of which combine to raise the efficiency above the best attainable in marine practice.

20. In an article of such limited length it is only possible at most to lay the foundation for a rational theory of propeller design. The details of its application must therefore be left for future communications. It seems desirable, however, to add a few notes as to the nature of the quantities involved in these formulas.

In practice the torque, M, and revolutions per minute, N, are determined by the power plant installed. The propeller must be so designed, therefore, as to develop the specified torque, M, at the specified speed of revolution, N.

The diameter, D, has a fixed upper limit determined by the dimensions of the propeller-well in the case of single screw ships, or by hull clearance for multiple screws. With the present tendency to higher speed engines without change in hull form, the upper limit for D is often fixed by the speed of revolution rather than tip clearance. Consequently D may usually be regarded as one of the variables in the problem.

The speed, V, represents the speed of the vessel corrected for wake. This is fixed by the shaft horsepower and the form of afterbody of the vessel. For standardized hull types it is possible, in general, to estimate very closely the speed of the vessel for the shaft horsepower specified. The question of wake is more difficult. Joukowsky has thrown much light on the nature of wake by his work on associated vortices. For practical purposes, however, it is sufficient to establish a relation between effective wake and relative dimensions of propeller and afterbody of vessel. For single screw ships such an empirical relation has been established by the writer which gives the wake percentage within 2 per cent or less. The investigation on which this is based shows that there is no apparent relation between wake and block coefficient, as the form of afterbody alone is of any practical importance. The determination of ship speed and wake percentage then fixes the value of V. The writer's experience leads him to believe that the

value of V so determined is far more nearly correct than the value of the shaft horsepower specified, over which the designer has no control.

In the non-dimensional product $\frac{DN}{V}$ then, the only variable over which we have any control is D.

As mentioned in Article 13, the angle $\beta$ lies within restricted limits, say 2 degrees and 8 degrees. For the sake of efficiency it is desirable to make $\beta$ small, whereas to meet the requirements of thrust, it must be of a certain definite amount, depending on the load on the propeller. The gain in efficiency with lightly loaded propellers is therefore evident.

The non-dimensional shape factor, S, determines the blade profile so that the circulation shall vary uniformly along the blade, decreasing to zero at the tip. Its numerical value is determined by the load on the propeller. The factor, B, occurring in S is dimensional, and determines the blade width for assigned values of D and S.

The quantities $K_1$ and $K_2$ are the only factors in any sense empirical. Such empirical factors are essential to take care of frictional effects, blade interference, etc., which must be determined experimentally.

The quantity G is a physical constant depending on the density. It has therefore one value for fresh water and a different value for salt water.

The parameter $\lambda$ lies very close to unity. Since it always appears in the product form $\lambda\beta$, we may therefore assume its value to be unity without loss of generality since the choice of $\beta$ is at our disposal.

21. These formulas are evidently well adapted to slide rule computation. Their use in design may be still further facilitated by presenting them in the form of alignment or nomographic charts, which eliminate all the labor of computation and consequent numerical errors.

A unique feature of these formulas is that they involve neither pitch nor slip. To compare results based on these formulas with those expressed in terms of pitch and slip, it is convenient to choose a suitable radius for comparison. The point chosen for this purpose is usually at .7 of the tip radius although the writer

uses a slightly different value. Then if $s$ denotes the true slip as related to a constant pitch wheel; if $p$ denotes the pitch of the blade at this radius, and if $q$ denotes the ratio of this pitch to the diameter, that is, $q$ is the pitch ratio $q = \frac{p}{D}$, then it is a simple matter to establish the relation

$$\frac{V}{ND} = \frac{q\,(1 - s)}{60} \tag{100}$$

This relation makes it possible to compare the performance of any propeller designed in accordance with these formulas with a conventional propeller of the so-called "true screw" type. It also makes available all the voluminous data obtained from model tank experiments in various countries, for determining the values of the empirical factors $K_1$ and $K_2$.

22. In any branch of engineering, the ultimate aim, of course, is to put it on a strictly rational and scientific basis. Hydrodynamics, by reason of certain difficulties which it presents, has been slow to yield to rational treatment. Marine propulsion was the first branch of hydraulic engineering to receive attention, but has remained wholly empirical. This is also true of more recent hydrodynamic applications such as centrifugal pumps, and steam, gas and hydraulic turbines. Evidently then we are considerably nearer our goal when we have established a rational basis for treating the phenomena of fluid flow and applying it to a specific problem, as in the case under consideration. From an engineering standpoint, this is the most important aspect of the present discussion.

The immediate consequence of rational methods of treatment is that we are enabled to handle all types of propulsive problems with equal facility, and with greater intelligence and certainty as to results. The reliability of a method absolutely sound in principle far more than offsets the experience of any particular individual, and represents a long stride in transforming propeller design and propulsive problems in general from an art to an exact science. This is the practical aspect of the present discussion.

To obtain a high value of propulsive efficiency, the first essential is to secure the most favorable combination of the primary variables involved, such as diameter and blade area. The advantage

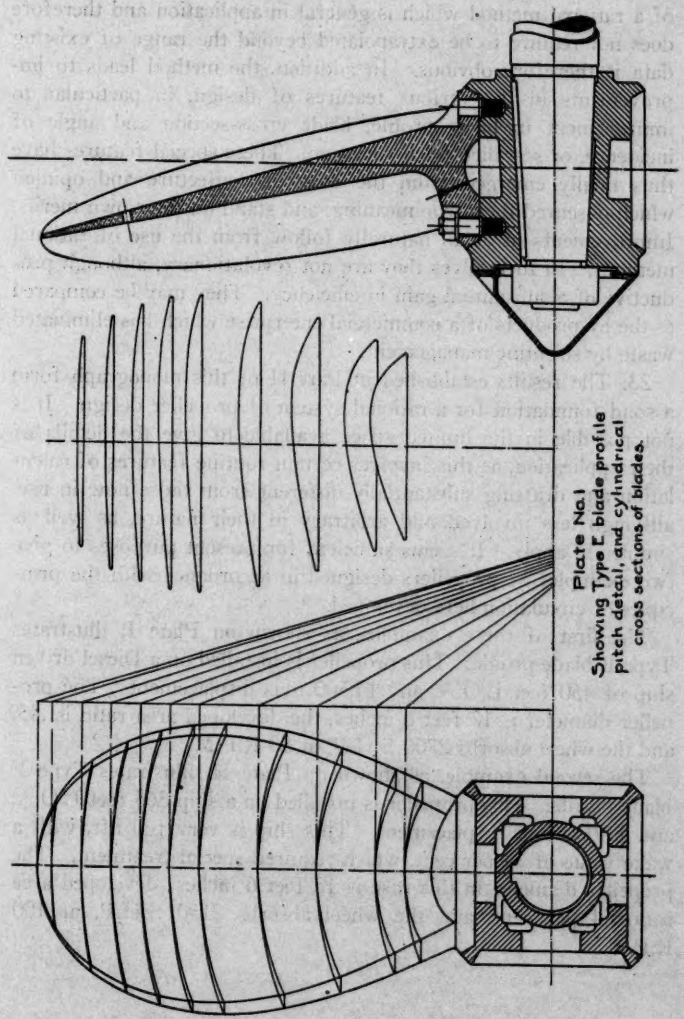

Plate No.I
Showing Type E. blade profile pitch detail, and cylindrical cross sections of blades.

of a rational method which is general in application and therefore does not require to be extrapolated beyond the range of existing data is therefore obvious. In addition, the method leads to improvements in the various features of design, in particular to improvement in blade profile, blade cross-section and angle of incidence, or so-called pitch variation. These special features have thus finally emerged from the haze of conjecture and opinion which obscured their true meaning, and stand on their own merits. Improvements in detail naturally follow from the use of rational methods. In themselves they are not revolutionary, although productive of a substantial gain in efficiency. They may be compared to the by-products of a commercial enterprise which has eliminated waste by scientific management.

23. The results established in Part II of this monograph form a solid foundation for a rational system of propeller design. It is not possible in the limited space available to give the details of their application, as this involves certain routine features of calculation and drafting substantially different from those now in use, although less involved and arbitrary in their nature, as well as simpler to apply. It seems sufficient for present purposes to give two examples of propellers designed in accordance with the principles of circulation here developed.

The first of these examples, as shown on Plate I, illustrates Type E blade profile. This propeller is installed on a Diesel driven ship of 450 feet L.O.A. and 14,500 tons displacement. The propeller diameter is 17 feet 6 inches, the developed area ratio is .45, and the wheel absorbs 2700 S.H.P. at 80 R.P.M.

The second example, as shown on Plate II, illustrates Type F blade profile. This propeller is installed on a ship 365 feet L.O.A. and 11,000 tons displacement. This ship is very full aft, with a wake value of 54 per cent, which required special treatment. The propeller diameter in this case is 15 feet 6 inches; developed area ratio 54 per cent, and the wheel absorbs 2150 S.H.P. at 100 R.P.M.

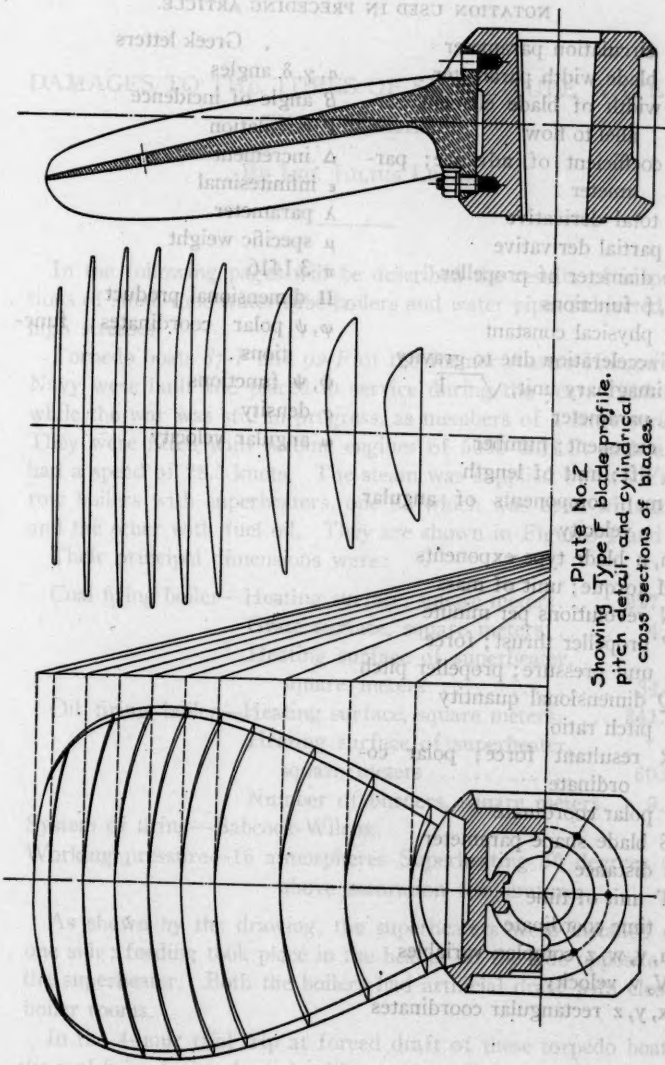

Plate No. 2
Showing Type F blade profile, pitch detail, and cylindrical cross sections of blades.

NOTATION USED IN PRECEDING ARTICLE.

A circulation parameter
B blade width parameter
b width of blade element parallel to flow
c coefficient of advance; parameter
d total derivative
$\partial$ partial derivative
D diameter of propeller
F, f functions
G physical constant
g acceleration due to gravity
i imaginary unit, $\sqrt{-1}$
K parameter
k exponent; number
L lift; unit of length
l, m, n components of angular velocity
m, n blade type exponents
M torque; unit of mass
N revolutions per minute
P propeller thrust; force
p unit pressure; propeller pitch
Q dimensional quantity
q pitch ratio
R resultant force; polar coordinate
r polar coordinate
S blade shape parameter
s distance
T unit of time
t time coordinate
u, v, w, z complex variables
V, v velocity
x, y, z rectangular coordinates

Greek letters

$\alpha, \gamma, \delta$ angles
$\beta$ angle of incidence
$\Gamma$ circulation
$\Delta$ increment
$\epsilon$ infinitesimal
$\lambda$ parameter
$\mu$ specific weight
$\pi$ 3.1416
$\Pi$ dimensional product
$\varphi, \psi$ polar coordinates; functions
$\Phi, \Psi$ functions
$\sigma$ density
$\omega$ angular velocity

# DAMAGES TO THE TUBES OF SMALL-TUBE WATER-TUBE BOILERS.

By Ing. Julius Levai.

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In the following pages will be described the results of inspections of small-tube water-tube boilers and water pipes subjected to high stresses.

Torpedo boats *87-F* and *92-F* of the former Austro-Hungarian Navy were built and placed in service during the years 1915-16, while the war was still in progress, as members of a large series. They were fitted with turbine engines of 5000 B.H.P. each and had a speed of 28.5 knots. The steam was supplied by two Yarrow boilers with superheaters, one of which was fired with coal and the other with fuel oil. They are shown in Figures 1 and 2.

Their principal dimensions were:

Coal firing boiler—Heating surface, square meters..... 227.6
Grate surface, square meters..... 4.89
Heating surface of superheater, square meters ..... 34.6
Oil firing boiler—Heating surface, square meters..... 341.5
Heating surface of superheater, square meters ..... 60.2
Number of burners, square meters. 9.
System of firing—Babcock-Wilcox.
Working pressure—16 atmospheres Superheating 50 degrees C., above saturation temperature.

As shown by the drawing, the superheaters were installed on one side; feeding took place in the header on the side opposite to the superheater. Both the boilers had artificial draft with closed boiler rooms.

In the 4-hour trial trip at forced draft of these torpedo boats, the coal-firing boiler showed, with a consumption of 1900 kilograms

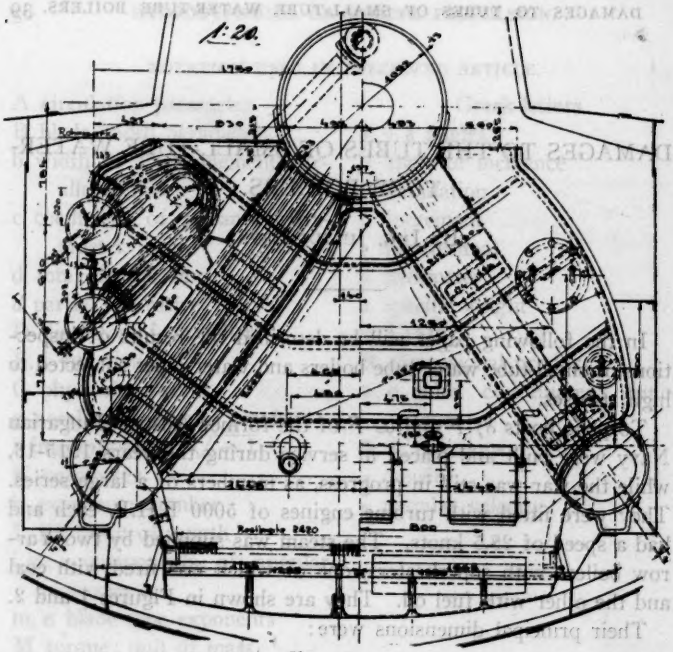



FIGURE 1.

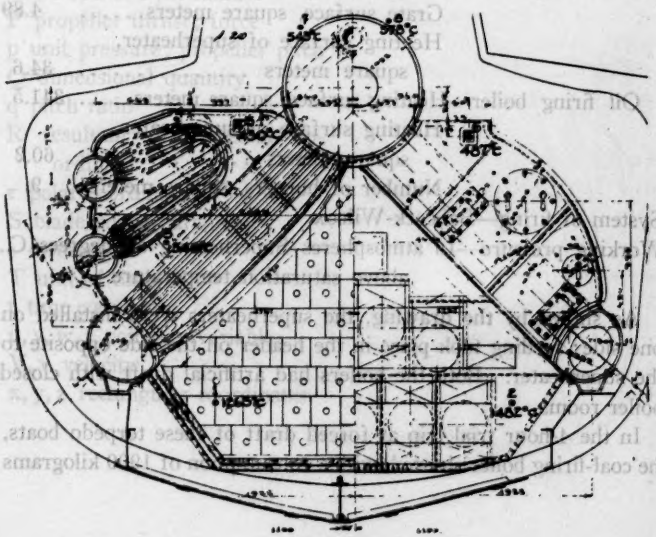
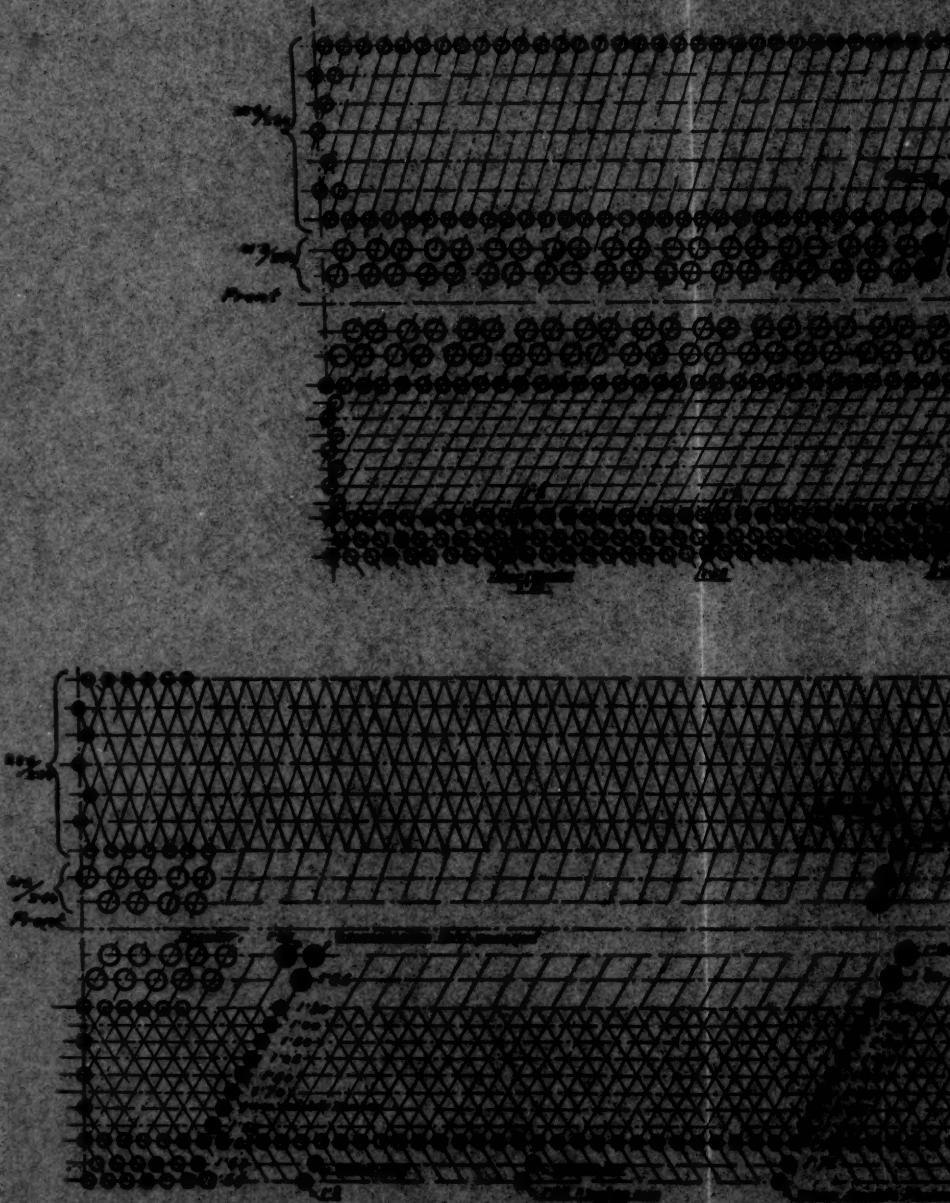
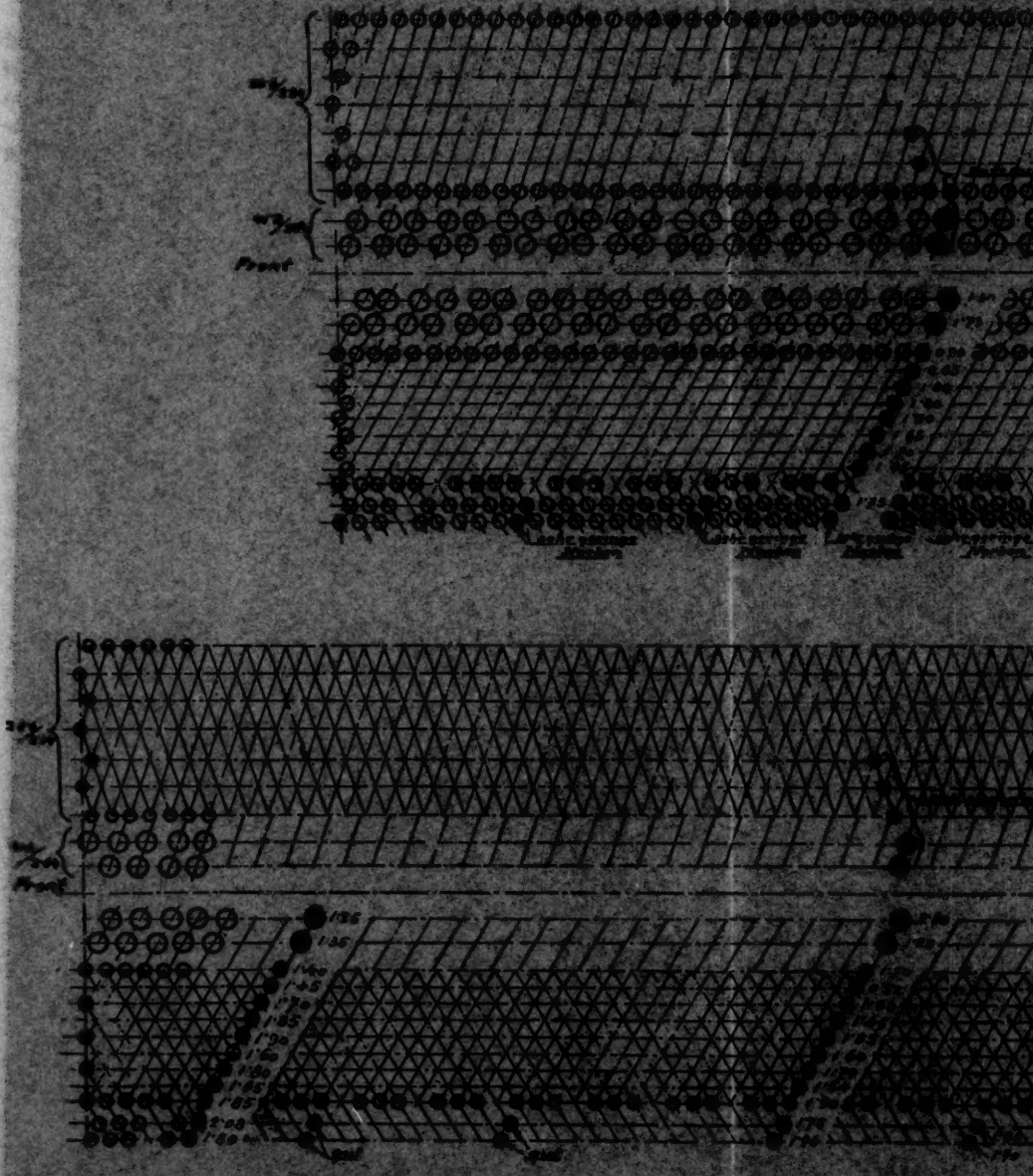
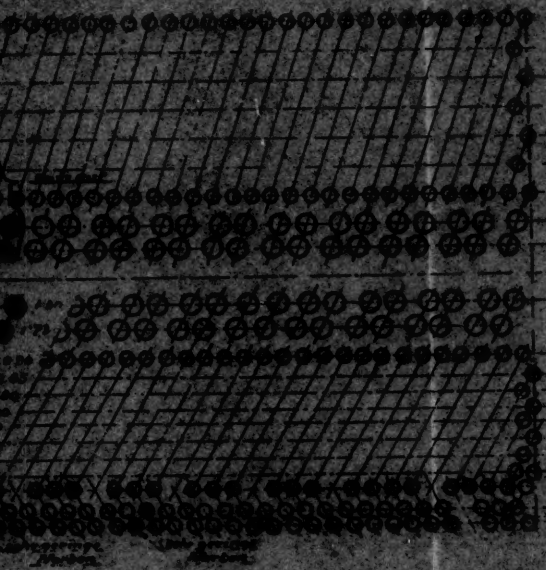
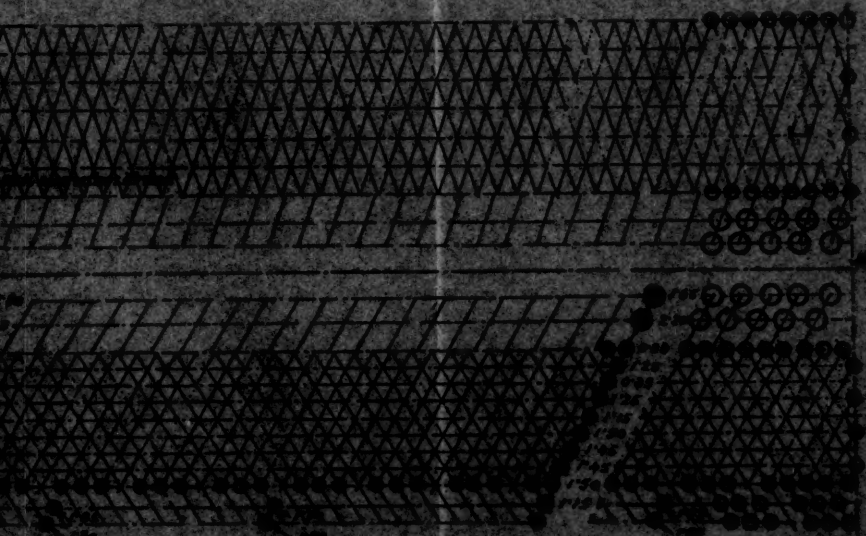



FIGURE 2.









Nixon coal per hour, heating value 8200 kilogram-calories, 13,500 kilograms steam per hour; the oil-firing boiler showed with a consumption of 2665 kilograms Galician fuel oil per hour, heating value 11,000 kilogram-calories, 26,500 steam per hour; the load in the coal-firing boiler was therefore

388 kilograms coal per square meter of grate surface, and
59.3 kilograms steam per square meter heating surface; in the oil fired boiler
7.8 kilograms fuel oil per square meter heating surface, and
77.3 kilograms steam per square meter heating surface.

Torpedo boat *87-F* was completed in September, 1915, *92-F* in March, 1916, and both were at once placed in service.

At the end of about 5 months operation leaking tubes were discovered in the innermost row of tubes on the feed side, the leakage being caused by small circular pits passing through the wall of the tube and ending on the outer side in holes as big as the head of a pin. These tubes were changed while the vessels were still in service but as other tubes began to leak in the course of time, both torpedo boats had to report at the shipbuilding yard, about the middle of December, 1916, that is, after scarcely 14 and 8 months' service, respectively, for a thorough inspection. The startling rapidity with which the tubes had deteriorated made it necessary to give the boilers a thorough examination, with the following results:

On both torpedo boats entire transverse rows of tubes were removed from the nests on both sides of both boilers in their entire length; then at one place on the circumference others were cut open throughout their entire length and laid flat. They were pickled in highly diluted hydrochloric acid until entirely freed from their inner coating, and were kept in this condition by the use of oil, so that the erosions on their inner surface could be plainly seen and the thickness of the walls in these erosions easily measured. In each torpedo boat 27 tubes were thus removed from each of the coal-firing boilers and 56 from each of the oil-firing boilers and so treated in preparation for the investigation. The tubes removed in this way are shown in black in Figures 3 and 4. The thickness of the wall where the pitting was deepest—that is to

say the minimum thickness found—is noted alongside each tube that was removed.

The character of the inner surface of the tubes is shown in the photograph, Figure 5.

The pitting was distributed lengthwise of the tubes with such astonishing regularity that, as a matter of interest, it has also been indicated on a diagrammatic sketch of the boilers, reproduced here as Figures 6 and 7, which can likewise be used as distribution diagram. In these diagrams the eroded portions of the tubes are shown in black. The pits were always more numerous on the side of the tube turned towards the fire.

These measurements and notations furnish a general idea of the nature of the erosions, namely:

1. That the damages in all four cases appeared to have been produced by similar causes;

2. That the deepest erosions were found in the innermost row of tubes turned towards the fire; from there on they gradually decreased and from the 5th to 6th rows on were considered, as a rule, not of a nature to cause anxiety. The tubes of the next to the inside row of large tubes showed the deepest and most numerous erosions while the two outermost rows of feed tubes were the least damaged. In the vicinity of the front end of the boiler, the two rows last mentioned showed almost no erosion. Towards the rear wall, where less feed-water got into the tubes, the pits increased in depth and number.

3. The same is true also as regards the extent of the erosions. The damaged part of the tube—which was always the lower part—likewise always decreased in length the further it was from the fire, so that the two outermost rows of tubes were damaged only in a slight degree or not at all.

The photographs reproduced in Figure 5 show respectively the damages to a tube of the second (large), the fourth and the thirteenth rows on the side of the superheater, counting from the inside. The individual deep erosions, enclosed in a circle on the photographs, are very striking; such erosions were also found in tubes not otherwise attacked and made it necessary to change all the tubes on the feed side.

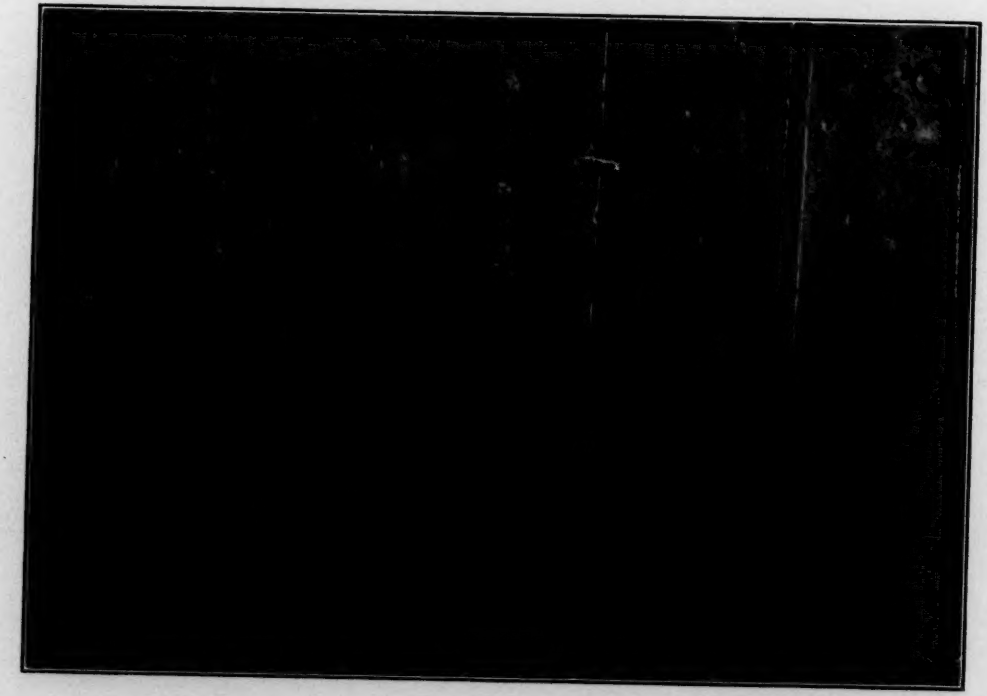


FIGURE 5.

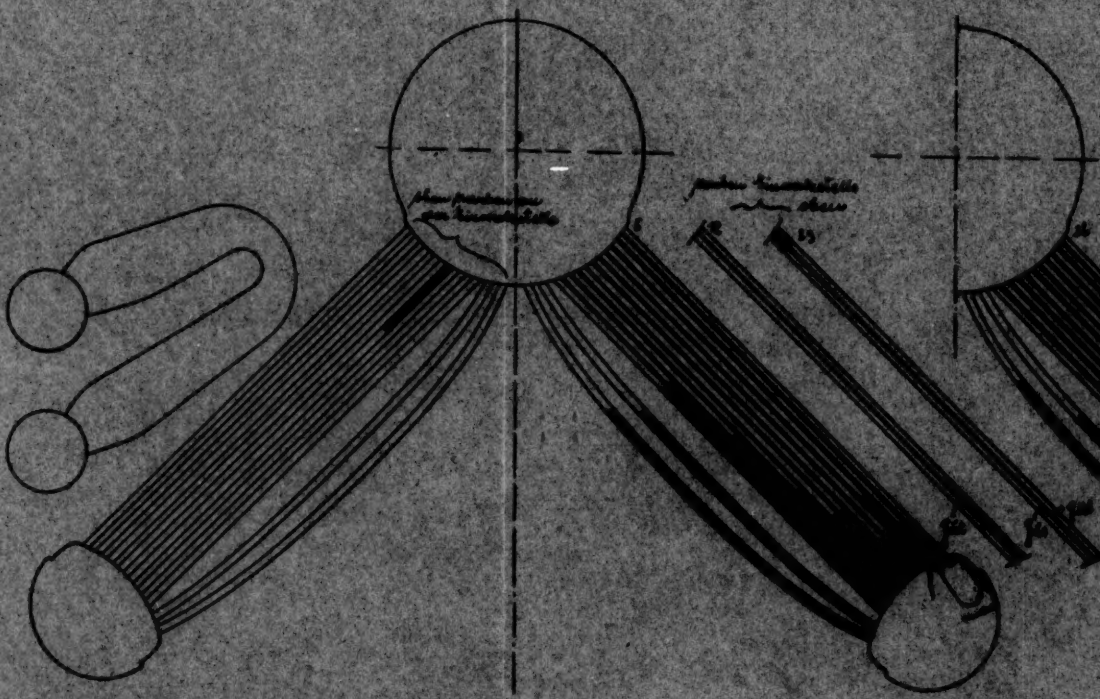
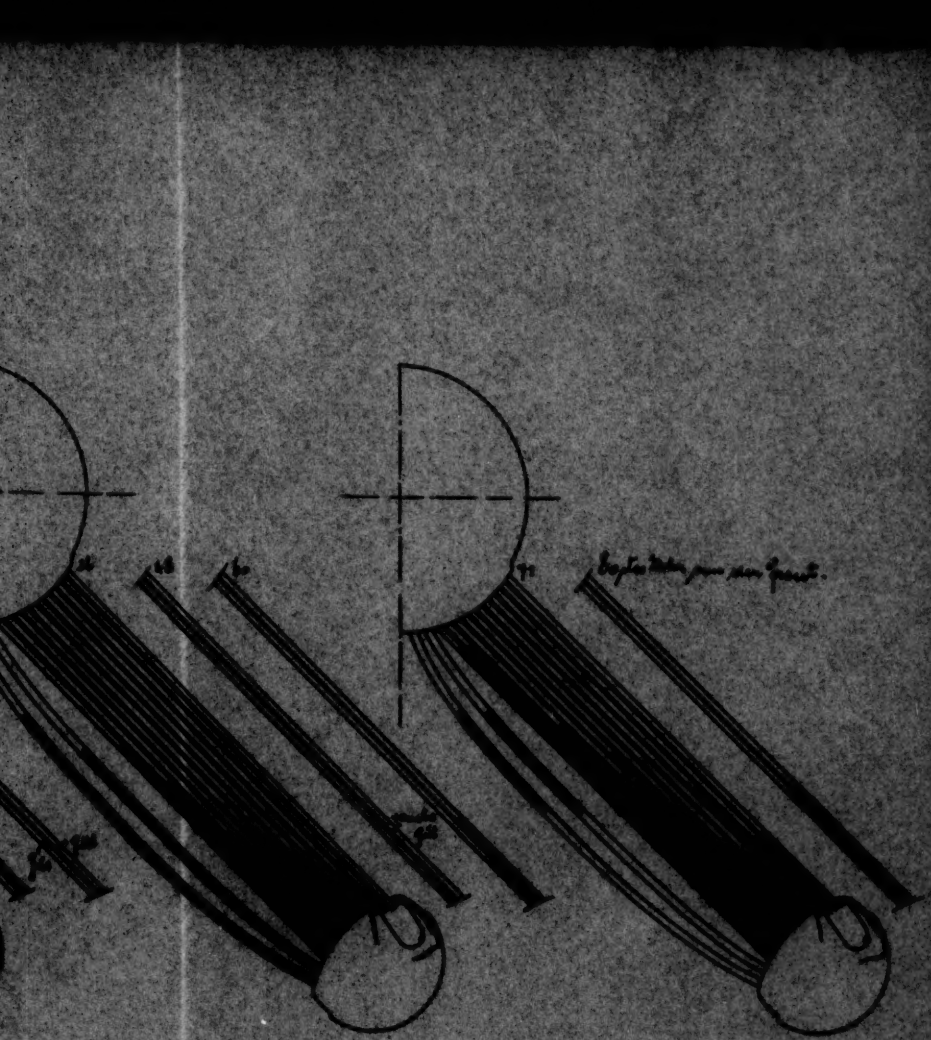


FIGURE 6-2.



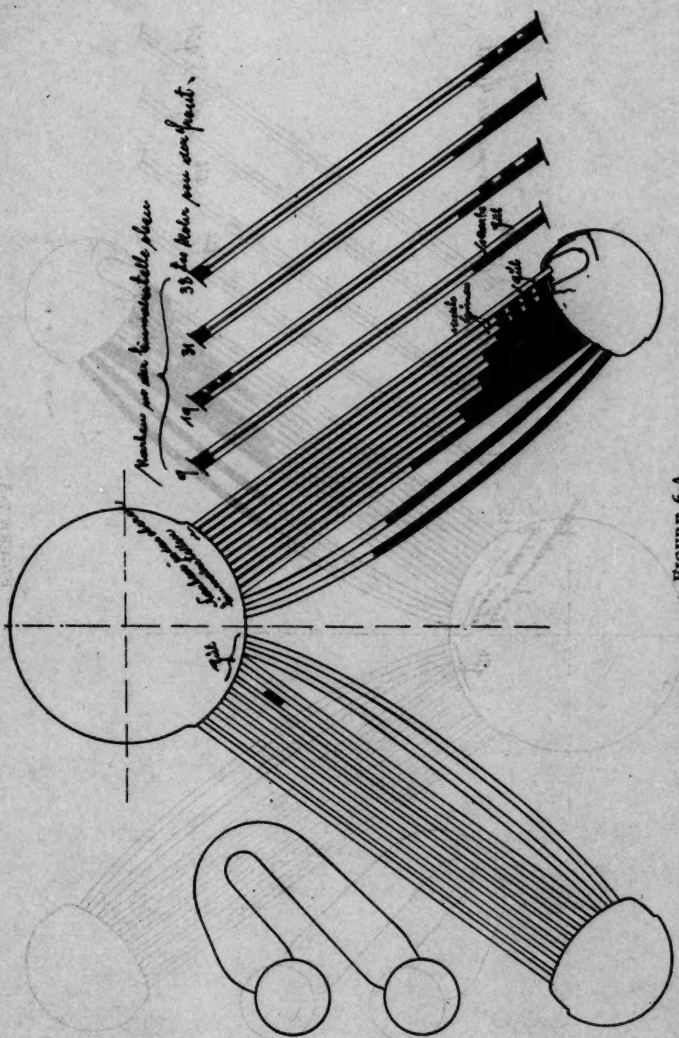


FIGURE 6-A.

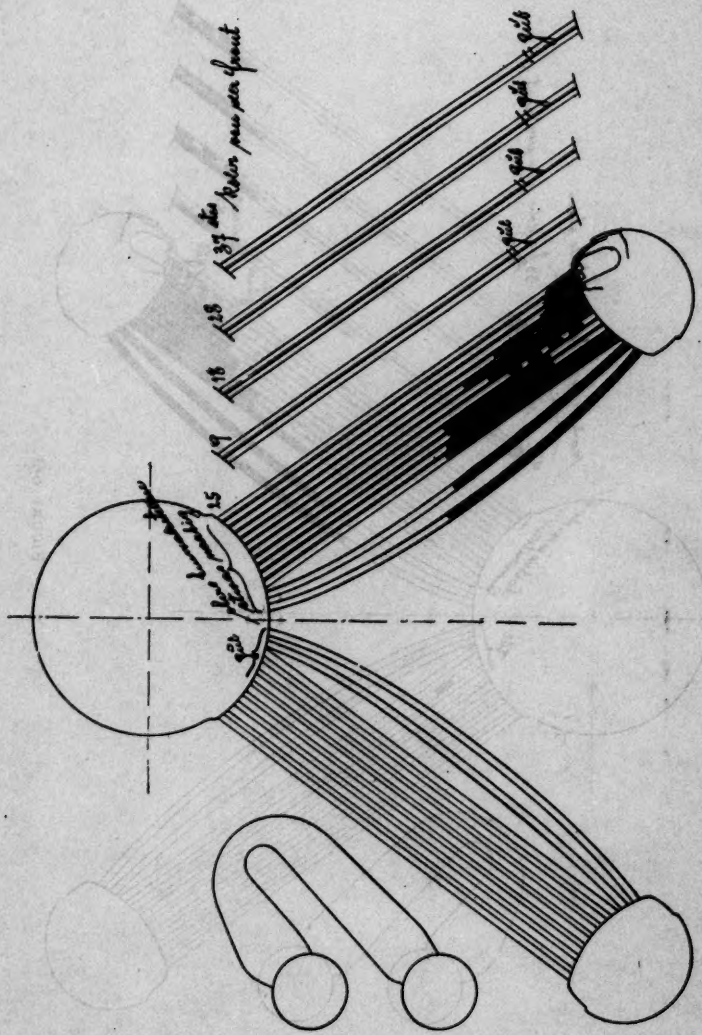


FIGURE 7-A.

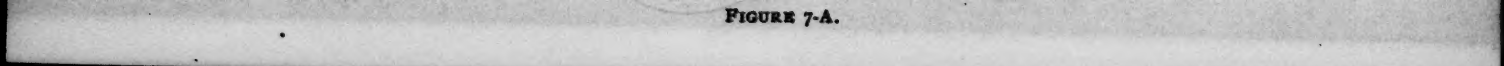


FIGURE 7-A.

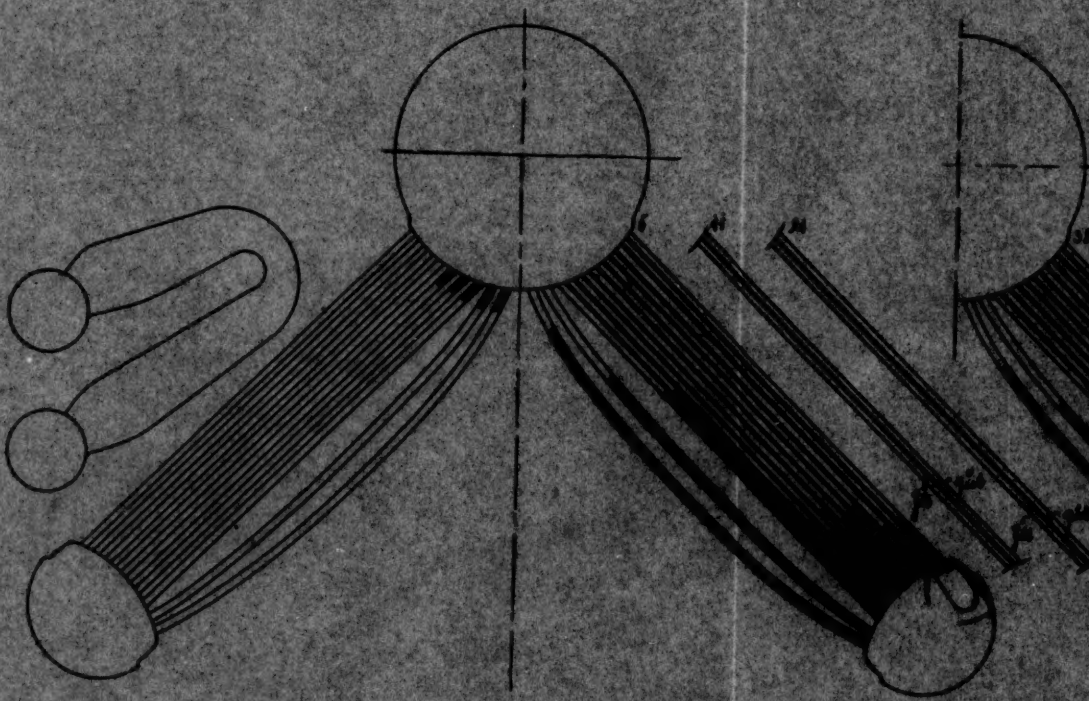


FIGURE 7-2.

The following may be surmised as to the cause of the trouble:

If we disregard the leakage in the feed-water supply piping in the header, the cold feed-water only reaches the two outermost rows of tubes on the feed side, is there preheated and, as is always the case in the water circulation in the boiler, does not flow into the other tubes until it has passed through the steam collector. All these tubes thus receive water which has first been freed of air in the steam collector. Hence if the cause of the damages is to be sought in the air content of the feed-water, the two outermost rows of tubes, and only these, would in every case be attacked. As this was not found to be the case in any of the four boilers examined, the air had to be discarded as the cause of the damages.

The fact that the steam collector showed no erosions, and the astonishingly short time in which erosions entirely penetrating the wall were produced, likewise argued against the acceptance of the air as the cause of the trouble.

The further examination of the boilers disclosed a thin and very adhesive inner coating which was considerably more fatty on the feed side than on the superheater side. This also agrees with observations made in connection with other investigations of this kind. The coating could be removed from the tubes only with a sharp instrument and the application of force and was similar in appearance to forge scale. This coating likewise constituted an obstacle to the direct adhesion of the air to the inner walls of the tubes.

This coating, as well as the boiler water, showed a marked salt content, not more, however, than is customary in sea-going vessels with surface condensation.

It seems quite reasonable to suppose that the extremely rapid destruction of the water tubes on the feed side is to be attributed to grease.

This supposition is strengthened by the following: In normal operation a two-fold change in direction of the feed-water takes place on the feed side, and a single change on the superheater side, as shown in Figure 8.

The water flowing through the 12th and 13th rows of feed tubes and over the baffle plate into the steam collector on the

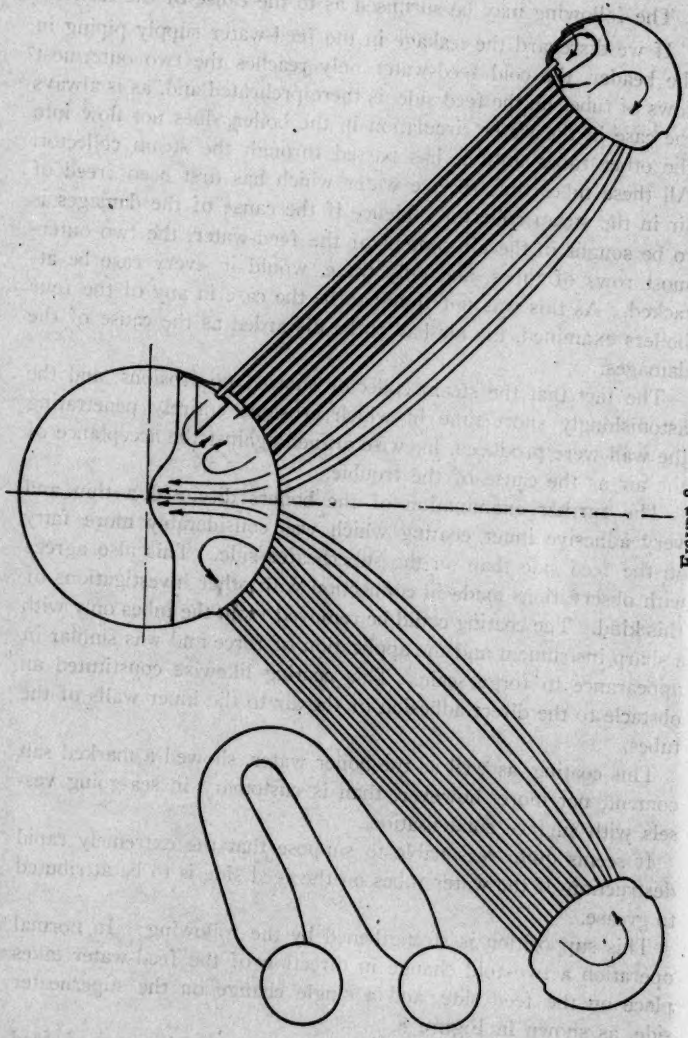


FIGURE 8.

right is carried along with the prevailing current into the adjacent rows of small tubes on the feed side; this is all the more true because the steam and water mixture flowing forcibly outward at the center to the inner rows of tubes on both sides, prevents the direct flow of the water fed in on the right side over to the left side of the steam collector. The tubes of the superheater side therefore receive chiefly water of this kind which has left the feed side after a two-fold change of direction and has meanwhile already deposited most of the grease. This might well be the cause of the heavy coating of grease on the feed side.

Fuel oil got into the water of condensation through leakage of the fuel-oil preheater, which was heated by steam. With leaking heating tubes in the fuel-oil heater, the fuel oil was carried along into the condenser piping or into the condenser itself, from whence the feed water carried the grease into the boilers.

The astonishing rapidity of the erosion is also to be attributed to the high temperatures which occurred in the customary forcing of the boilers during the war. A measurement of the temperatures is shown here in Figure 2.

FIGURE 8.

These measurements were taken during two 4-hour runs. The measurements taken at points 1 and 2 gave the temperature in the furnace, platinum-rhodium pyrometers being used. Those taken at 3 showed the temperature of the smoke-gas in front of the superheater, 5, 6 and 7 in the smoke-flues, 8 and 9 in the funnel; all these measuring points were fitted with *"Eisen-Konstantan"* pyrometers.

The temperatures are shown on the boiler diagrams. When we think that the flames come in direct contact with the innermost rows of tubes, it is easy to understand that the iron was unable to resist the acids produced from the oil at temperatures of about 1000 degrees-1200 degrees C.

A further typical analogous case of the erosion of water tubes, in which, however, the manifestation of the damage is somewhat different, occurred in coast-defense torpedo boats of 110 tons displacement fitted with engines of about 2400 H.P. and two boilers of the White-Foster type with bent tubes, for oil firing, but without superheater. The boilers had artificial draft with closed boiler rooms.

The principal data for these boilers were, for each boiler:

Heating surfaces—203 square meters.
Number of burners—5.
System of oil firing—Babcock-Wilcox.
Working pressure, wet steam—19.2 atmospheres.
Oil burned per hour—912 kilograms.
Fuel-oil per square meter heating surface per hour—4.5 kilograms.

This typical case is shown in Figure 9 in which two tubes, treated as described at the beginning of the present article, are illustrated.

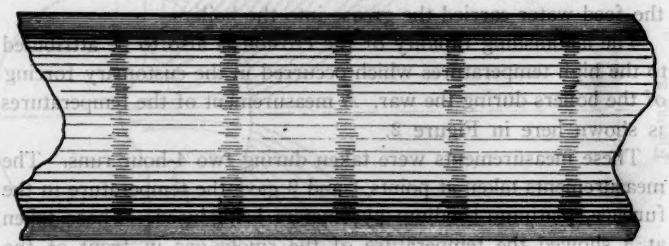


FIGURE 9.

I had already observed similar damages in other boilers, where they became apparent only after pickling in diluted hydrochloric acid. The erosions always occurred on the inner surface of the tube in the form of parallel grooves lying close together side by side and perpendicular to the longitudinal axis of the tube; in some places the erosion extended entirely through the wall and caused the tubes to leak. In these cases also the damage could be traced only to the action of the fuel oil, which made its way into the drainage system and from there into the condenser, and finally into the boilers through leakage of the fuel preheater. The fuel oil was deposited on the inner surface of the tubes and contributed to the erosion of the material. The expansion and con-

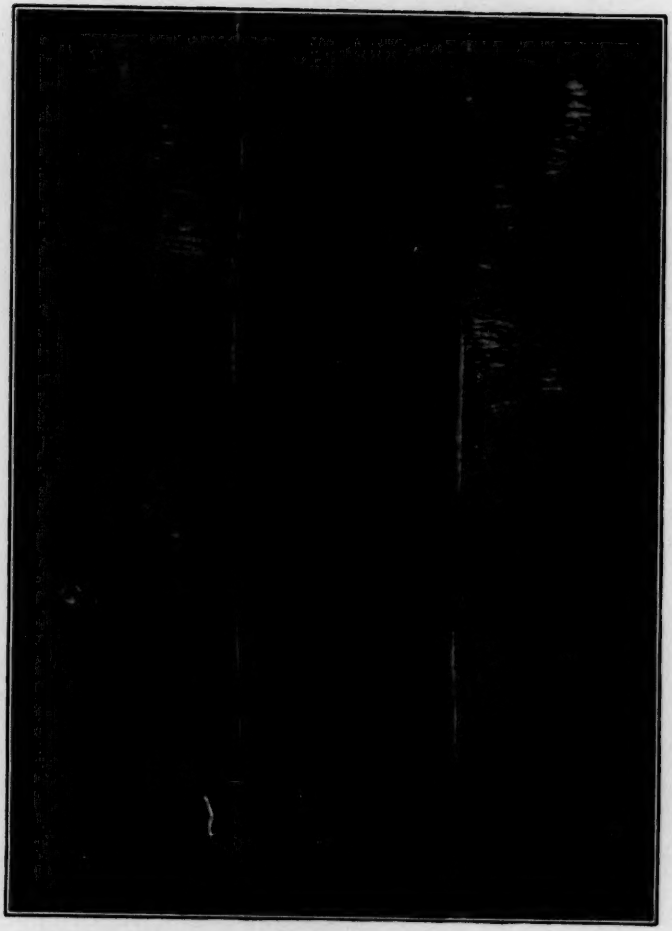


FIGURE 10.

traction, or the bending of the tubes due to the heat in operation likewise probably plays a not inconsiderable part among the more detailed causes of this erosion. Even though the tubes can expand with almost perfect freedom between the steam collector and header, the longitudinal expansion of the tubes will be controlled by their mean temperature. On the other hand, in boilers with oil firing, the tubes of the innermost row expand according to the maximum absorption of heat through radiation and conduction, especially when the flames come in contact with them, to a very much greater degree than the rest of the tubes, and are forced to make room for themselves by bending. I had often observed in boilers under high load how the tubes of the innermost rows bent to such an extent, with the turning on and off of the burners, that is, through alternate heating, that the middle part in a few minutes projected beyond the rest of the series and then as quickly withdrew, the rise of the curvature thus produced amounting to as much as one and one-half times the diameter of the tube. The bending stresses thus produced transversely to the axis of the tube in conjunction with the high temperature in part explain the damages shown in Figure 9, provided the presence of oil or grease also contributed.

The peculiar nature of the damage cannot be fully explained, however, without taking the process of manufacture into account.

The tubes were made according to the Mannesmann-Pilger process, in which, according to the specifications, they were drawn through the passes while cold. By this process the diameter of the tube is reduced to a smaller one, during which, especially if the pass has become defective from long use, the operation of drawing is associated with marked vibrations in consequence of which the finished tube shows annular irregularities fissures on its inner surface. See Figure 10. These irregularities can best be seen by illuminating the inside of the tubes immediately after drawing, while they are still quite bright. The continuity of the iron is no longer perfect at such places, the texture of the material is curly, and if the curvatures described above should occur, it is easy to see how the inner wall of the tube succumbs to chemical action in the manner shown in Figure 9.

The tube shown in Figure 9 was subjected to a microscopic-chemical examination. The tube from the innermost row was cut into rings and longitudinal strips and the pieces were examined. Nine photographs were taken, which have been reproduced in Figure 11.

Photograph "a" shows at the right the outside of the tube, on the side turned towards the fire. It is strongly oxydized, burned in places on the surface and coated slightly with a white lime deposit. The middle piece shows the other side of the tube, that is, the side turned away from the flame, which is only slightly coated with a greasy, sooty, black deposit. The figure at the left shows a section of the tube which was so carefully made that the inner coating was not disturbed. On the inside of the tube at the lower part may be seen a coating more than a millimeter thick, corresponding to the side turned towards the flame. This coating formed a hard crust consisting of ordinary salts such as form boiler scale, and some fat. Beneath this thick coating were the peculiar parallel groove-shaped erosions running transversely to the tube; all these erosions were originally filled with scale. The upper side of the annular section was coated and eroded only to a very small degree. Longitudinal and cross sections on an enlarged scale are shown in photograph "b."

The character of the erosions may be seen still more plainly in the micro-photographs "c" and "d," which represent sections through the inner surface of the side of the tube exposed to the fire, and "e," which shows a section through the opposite inner side of the tube. These photographs show that the erosions have no connection with the internal texture of the material, and they also show the great difference in the extent of the erosion. Photographs "f" and "g" show cross-sections of the tube still further magnified, "f" showing the outside and "g" the inside surface of the side turned towards the flame.

The difference in the size of the ferric-crystals, which are many times larger on the inside than on the outside, is a striking feature. This difference does not occur on the side turned away from the flames. These large crystals can be explained only by the assumption that at times the material was heated above 800 degrees C. and then gradually cooled off while the wall of the tube on the

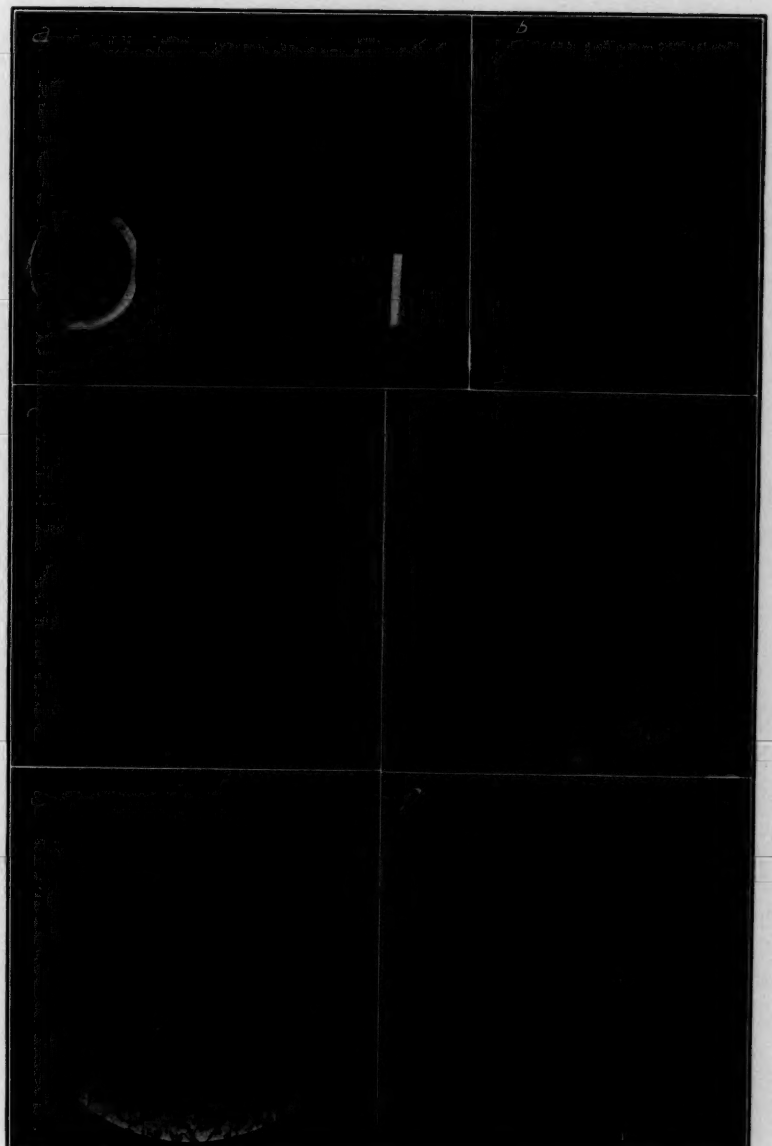


FIGURE 11.

FIGURE 11.

water side was protected against rapid loss of heat by the inner coating. That this alternate heating and cooling must have been frequently repeated is indicated by the slight sensitiveness of the crystals to the action of costic fluids and the peculiar configurations of the erosions which are especially pronounced on the ferric-crystals in contact with boiler scale. This type of erosions, which has nothing to do with the internal crystalline structure, is an indication of fatigue of the metal. They can be plainly seen in photographs "h" and "i."

# INTRODUCTION TO DYSON'S METHOD OF PROPELLER DESIGN.

By Commander J. M. Irish, U. S. Navy, Member.

1. For the past twenty years all of the propellers installed on Naval vessels have been designed by methods originated by Admiral Dyson, or if supplied by outside contractors they have been analyzed by the same means before adopted. In spite of numerous articles on the subject and three editions of his book on Propeller Design, comparatively few officers know the principles on which Admiral Dyson's design is based, and fewer still are capable of using his method for either analysis or actual design.

2. This attitude of mind is partly due to the impression that propeller action is a deeply mysterious thing, and secondly to the lack of some preparation which is necessary for an easy comprehension of the facts as presented in Admiral Dyson's books. The student will not find there any concise explanation of what Dyson's methods consist; nor is there given any summary of the steps involved and their relation to the whole scheme of design. As many of the problems are examples of how to treat special or exceptional cases, the beginner finds himself floundering along unable to see why the rules recently given are not followed, and soon gets the idea that either the rules do not hold, or else everything is an exception. This is not the case, however, for, as the curves were developed by analysis of the performance of numerous ships, they represent the conditions found on the great majority of naval ships. The exceptions are those of special hull or propeller forms and including them in the book was to demonstrate the feasibility of extending the application of the principles when skilfully used, or to show the effect of poor design.

3. In developing his method Admiral Dyson based his primary assumptions on the best theories of propeller design, but, as in applying all theories to practice, empirical factors are a large and necessary part of the final results. The basis of the method is the theoretically perfect performance of a given propeller when operating behind a hypothetical hull and absorbing the power at maximum possible efficiency under those conditions. The power, speed, and propeller slip so determined are designated as the "Basic conditions". The problem then resolves itself into determining the factors to be applied to the Basic conditions so as to permit calculating the power, speed and apparent slip of that same propeller when absorbing the actual horsepower and operating behind the actual vessel. The Dyson method consists of three major steps:

I. Calculation of the Basic Conditions.

II. Determination of the Factors.

III. Application of the factors to the Basic Conditions to Give Actual Performance of the Vessel.

4. Each of these major steps have numerous subdivisions, and as the given problem may be one of design, or analysis, or both, the sequence of the various steps may be greatly varied depending on what are the knowns and the unknowns, but each of those steps must be gone through before the final answer is obtained.

5. As noted above the Basic Conditions are Basic Shaft (or Indicated) Horsepower, the Basic Effective Horsepower, the Basic Speed and the Basic Slip. The factors necessary are the Net Power Factor, designated as $Z_p$, the Speed Factor $Z_s$, and the Hull Factor K. This last factor, K, results in an increase in the power required for a given speed so Dyson calls it the power augment factor, but it is the combined results of the Hull lines and the location of the screw in reference to the hull. After K is determined it is combined with $Z_p$ in a manner described later, to form another Power factor $Z_g$, called the gross power factor.

6. Let us now consider how the Basic conditions are computed. Chart 20-A gives the theoretical performance of a propeller when operating at best efficiency and the data is based on the ratio of Projected area to Disk area ($Pa \div Da$). With a propeller of $Pa \div Da = .304$, for example, we would obtain the following:

T. S. = Tip speed = 6850

I. $T_D$ = Indicated thrust per square inch disk area = 3.8

P. C. = Propulsive coefficient = .678

$$\text{Then the Basic I. H. P.} = \frac{P \times T.S. \times I.T_D \times D \times N}{\pi \times C} \quad (1)$$

where P = Propeller Pitch in feet

D = Propeller Diameter in feet

N = Number of shafts

and C = 291.8 for 3 bladed propellers

or 252.4 for 4 bladed propellers.

$$\text{Basic S.H.P.} = \text{Basic I.H.P.} \times .92 \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$\text{E.H.P.} = \text{Basic I.H.P.} \times \text{P.C.} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

or Basic S.H.P. may be gotten directly by picking off $S.T_D$ which is shaft thrust per square inch disk area, and using it in place of $I.T_D$ in equation (1). Then Basic E.H.P. is gotten by picking out $P.C_s$ from upper propulsive coefficient curve of Sheet 20-A, in this case Basic E.H.P. = Basic S.H.P. $\times$ $P.C_s$ instead of equation (3).

$$\text{Basic Speed} = V = \frac{T.S. \times P \times (1 - S)}{\pi \times D \times 101.33} \quad . \quad . \quad . \quad (4)$$

Where T. S., P and D are as before and S = Basic Slip. This Basic Slip is determined by means of the hull dimensions and various charts,

Sheet 17-A determines the type of vessel

Sheet 17-B determines the Slip Block Coefficient, depending on the type.

Sheet 20-B determines the Basic Slip by means of Slip Block coefficients.

The use of these Sheets will be explained more fully in example below.

(B) Next consider the determination of factors.

7. (1) The net power factor $Z_p$ is obtained from the effective horsepower of the ship for a given speed and the Basic E.H.P. The e.h.p. is given in the curve obtained by the model tank for that hull, or it may be calculated by methods described in books on Naval Architecture, or Dyson, page 38. With the ratio

e.h.p. ÷ E.H.P. enter Sheet 21 and on curve (3) read off $Z_p$ from left hand scale.

(2) The speed factor is obtained by using ratio of speed of ship, "$v$", and basic speed, "V". This speed "$v$" should be the same as that for which the e.h.p. was picked out for the ratio e.h.p. ÷ E.H.P. This ties the two steps together, with $v$ ÷ V enter Sheet 21 and on curve (1) pick off $Z_s$ from left hand scale.

(3) K, the hull factor, is determined by use of Sheet 17-B which gives K Block Coefficients and Sheet 21-A which gives the value of K to be used based on the K block coefficient.

8. (C) *Application of Factors.*

The gross power factor $= Z_g = Z_p - \text{Log } k$ . . (5)

The estimated shaft horsepower of engine =

$$\text{S.H.P}_d = \frac{\text{Basic S.H.P.}}{10^{Z_g}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The apparent slip of vessel $= s =$ Basic Slip $\times$

$$\frac{10^{Z_s}}{10^{Z_g}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$\text{The estimated Revolutions} = R_d = \frac{101.33 \times v}{P \times (1-s)} \quad . \quad (8)$$

When $v$ = vessels speed in knots — same $v$ as was used in paragraph 7.
P = Propeller pitch
and $s$ = apparent slip as determined by equation 7.

9. To illustrate the use of the above formulae, let us take a simple example. Assume that a vessel has had a new propeller of slightly different dimensions installed, and that on leaving the yard there was only sufficient time for determining the revolutions for one speed over the measured mile. It is desired to construct the customary horsepower, speed and R. P. M. curves.

It is assumed that the vessel has a copy of its model tank curve of e.h.p. and that proper precautions were taken to insure that the I. H. P., speed and R. P. M. for that point are accurate.

We start first with the given hull dimensions:

Length on load water line = L = 520
Beam on load water line = B = 65.23
Draft for given displacement = H = 27.75
Displacement = 19452
Midship coefficient = M. S. Coef. = .984
Length of after body = L. A. B. = 183
Then B ÷ L = .125
H ÷ B = .425
$\frac{2 \text{ L.A.B.}}{\text{H}} = 13.2$

N. B. C. = Nominal Block Coefficient =

$$\frac{35 \times \text{Displacement in tons}}{\text{L} \times \text{B} \times \text{H}} = \frac{35 \times 19452}{520 \times 65.23 \times 27.75} = .725$$

We also have the given propeller data:

Twin screw, 3 bladed propeller
Projected area ratio = PA ÷ DA = .304
Diameter = D = 15'.954
Pitch = P = 14.422

The first step is the determination of the Basic conditions.

(I) *Basic Power.*

From Sheet 20-A for PA ÷ DA = .304
pick out T. S. = 6850
I. $T_D$ = 3.8
P. C. = .678

substitute in equation (1)

$$\text{Basic I. H. P.} = \frac{P \times \text{T. S.} \times \text{I. } T_D \times D \times N}{\pi \times C}$$

$$= \frac{14.422 \times 6850 \times 3.8 \times 15.954 \times 2}{\pi \times 291.8}$$

$$= 13100$$

Basic S. H. P. = I. H. P. × .92 = 12050
Basic E. H. P. = I. H. P. × P. C. = 8870

(II) *Basic Slip*, S, must be gotten before we can determine Basic speed, V.

Therefore:

(1) On Sheet 17-A plot M. S. coefficient .984 and H ÷ B = .425 and we find Hull is a Type 2.

(2) On Sheet 17-B, Chart I, Plot N. B. C = .725 and B ÷ L = .125 Call it point (*a*). Draw line from point (*a*) to lower right corner (1.0) of Chart. This line intersects line C-D at .593 read off base line. (C-D is used because it is a type 2 ship).

(3) On Sheet 17-A proceed now to Chart II and plot this value .593 and H ÷ B = .425. Call this point (*b*). Through point (*b*) draw curve parallel to curves shown until it intersects line C′ D′, read intersection off base line .60. (Chart II and line C′ D′ are used because it is a type 2 ship).

(4) Return with this value .60 to Chart I and plot .60 on line C-D. Call it point (*c*). Connect point (*c*) with right corner (1.0) and where this line cuts line X read .625 off base line. (Line X is used because it is a twin screw ship.)

This value .625 is called the Slip Block Coefficient.

(5) Turn to Sheet 20-B. Plot 2 L.A.B. ÷ H = 13.2 and Slip Block Coefficient .625 and read from left hand scale the Basic Slip = .195.

III. *Basic Speed* may now be obtained by substituting in equation (4).

$$\text{Basic speed in knots} = \frac{\text{T.S.} \times \text{P} \times (1-\text{S})}{\pi \times \text{D} \times 101.33}$$

$$= \frac{6850 \times 14.42 \times .805}{\pi \times 15.954 \times 101.33}$$

$$= 15.67 \text{ knots.}$$

We now have all the Basic Conditions and the next step is to determine the factors:

IV. *The Power Factor.*

From the e. h. p. curve of the vessel pick off the e. h. p. corresponding to the speed at which vessel ran across the measured mile. Let us say this speed was 14.7 and the corresponding e. h. p. is found to be 4420.

Then the ratio e. h. p. ÷ E. H. P. = 4420 ÷ 8870
= .498

On Sheet 21, using curve (3) we find that for this value .498 on the base line $Z_p$ the net power factor = .314 read off the left hand scale.

V. *The Speed Factor.*

We have the ratio $v$ ÷ V = 14.7 ÷ 15.67 = .94 and for this value on the base line on Sheet 21, using curve (1) we find the speed factor $Z_s$ = O

VI. *The Hull Factor* K.

(1) Referring back to paragraph 9—II where Basic Slip was found, on Sheet 17-A, Chart I, a point (*a*) was established by the N. B. C. and H ÷ B. The line from (*a*) to lower right corner also cuts line X at .618 when read off base line. This is the K. Block coefficient.

(2) Turn to Sheet 19 and erect a perpendicular from the base line at K. B. C. = .618. This cuts C-1 at diagonal line 1.21 which is the value of K. (C-1 is used because wing screws usually have large clearance from hull).

VII. As we have determined the net power factor and K, we can now obtain the gross power factor, $Z_g$ from equation (5)

$$Z_g = Z_p - \log K$$
$$= .314 - .0828 = .2312$$
$$\text{and } 10^{Z_g} = 1.70$$

VIII. With all these factors now established we have only to apply them.

$$\text{The estimated shaft horsepower} = \frac{\text{Basic S.H.P.}}{10^{Z_g}} \quad . \quad (6)$$

$$\text{S.H.P}_d = \frac{12050}{1.70}$$
$$= 7085$$

$$\text{The apparent slip} = \text{Basic Slip} \times \frac{10^{Z_s}}{10^{Z_g}} \quad . \quad . \quad . \quad . \quad (7)$$
$$= .195 \times \frac{1.0}{1.70}$$
$$= .1146$$
$$1-s = .8854$$

$$\text{The estimated revolutions} = R_d = \frac{101.33 \times v}{P \times (1-s)} \quad . \quad . \quad (8)$$
$$= \frac{101.33 \times 14.7}{14.42 \times .8834}$$
$$= 116.8$$

The actual ship on which this was done developed 7115 S.H.P. at 14.7 knots at 116.72 R.P.M.

In case the results do not agree closely with the observed, look for difference in displacement over that of the e. h. p. curve, or K may be slightly different and the computed K can be varied slightly so as to bring results closer together.

10. To construct the curve make up a table as follows—for several speeds pick off the slip corresponding and proceed for each speed as above described. Of course the Basic condition, power, speed and slip and factor all remain unchanged and do not have to be recalculated.

| | | | | | | |
|---|---|---|---|---|---|---|
| 1. Speed of vessel $= v =$ | 6.0 | 8.0 | 10.0 | 12.0 | 14.0 | 14.7 |
| 2. e.h.p. from curve | | | | | | 4420 |
| 3. e.h.p. $\div$ E.H.P. | | | | | | .498 |
| 4. $Z_s$ from curve (3) Sheet 21 for $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ | | | | | | .314 |
| 5. K same | 1.21 | 1.21 | 1.21 | 1.21 | 1.21 | 1.21 |
| 6. log $k$ | .0828 | .0828 | .0828 | .0828 | .0828 | .0828 |
| 7. $Z_g = Z_p - \log k$ | | | | | | .2312 |
| 8. $10^{Z_g}$ | | | | | | 1.70 |
| 9. $v \div$ V where V $= 15.67$ | | | | | | .94 |
| 10. $Z_s$ from curve (1) sheet 21 for $v \div$ V | | | | | | 0 |
| 11. $10^{Z_s}$ | | | | | | 1.0 |
| $s = s \times \frac{10^{Z_s}}{10^{Z_g}}$, where S $= .195$ | | | | | | .1146 |
| 1—S | | | | | | .8854 |
| $R_d = \frac{101.33 \times v}{P \times (1-s)}$ | | | | | | 116.8 |
| $S.H.P_d = \frac{S.H.P.}{10^{Z_g}}$, where S.H.P. $= 12050$. | | | | | | 7085 |

11. From such a table, which is an analysis, we could determine how efficient this propeller was. By assuming a smaller pitch we could analyze to see if we could get higher r.p.m. and more speed; or by reducing the diameter or the area (the PA $\div$ DA) we could determine if we could get the same speed on lesser horsepower.

11. *Cavitation.*

The methods of handling cavitation are covered in Dyson's book and are too extensive for this article, but a few points will help to follow his methods. Cavitation is the breaking down of the propeller action due to any one of a number of causes. It is evidenced by a decrease in efficiency of propulsion and is accompanied by an increase both of revolutions per minute and power over that which would be normally expected. The major causes are insufficient area for the thrust it is called upon to transmit or insufficient pitch causing speeding up of the propeller. Therefore Dyson calls the first type excess thrust cavitation and the other excess tip speed cavitation.

Both of these types of cavitation can be quickly and easily located by the Charts and the danger of unconsciously designing a screw in cavitation can be eliminated. Chart 22-A is used, and the curves are based on the e. h. p. ÷ E. H. P., the $v \div V$ and the Basic Slip.

To determine cavitation use the following rules.

*Rule 1* — to locate tip speed cavitation. Compute the ratio of ship's actual speed to Basic Speed ($v \div V$). If the ratio be found to be greater than 1.0, tip speed cavitation exists.

*Rule 2* — to locate excess thrust cavitation—Divide the Basic Shaft (or indicated) horsepower by the actual horsepower (shaft or indicated); this is $10^{Z_g}$. The log of $10^{Z_g}$ is $Z_g$. From curve (3) Sheet 21 pick off the corresponding e. h. p. ÷ E. H. P., this is e. h. $p_g$ ÷ E. H. P. Also compute the ratio $v \div V$. Plot these two on Sheet—using chart for the proper type of ship. If this point plots below the curve of Basic Slip, cavitation due to excess thrusts exists. If above this curve of S, cavitation does not exist.

Example (1) assume a type 1 ship, with Basic Slip of 12, Basic V 15, Basic S. H. P. 8000.

Actual speed $v = 16$.

$$v \div V = \frac{16}{15} = 1.06$$

Therefore excess tip speed cavitation exists.

Example (2) Assume same ship.

Actual $v = 10$ actual S. H. P. $= 7000$

(*a*) $v \div V = 10 \div 15 = .666$

$$10Z_g = \frac{\text{Basic S. H. P.}}{\text{Actual S. H. P.}} = \frac{8000}{7000} = 1.142$$

$$Z_g = .058$$

(*b*) e. h. $p_g \div$ E. H. P. $= .88$

Plot (*a*) and (*b*) on Chart II Sheet 22-A and as they plot below curve for S = 12. Cavitation exists due to excess thrust.

## DESIGN.

12. When commencing the design of a new vessel for the Navy certain preliminary work results in establishing the approximate displacement and the lines of the hull. From the model tank are obtained the effective horsepower curves for this hull over a wide range of speeds and for usually three different displacements. With this data it is the problem of the Bureau of Engineering to design a propeller and determine the horsepower of engines.

The designer is supplied with the following:

(*a*) Hull dimensions—length, breadth, draft and displacement.

(*b*) e. h. p. curves.

(*c*) General data regarding number of shafts, the approximate revolutions and the speed desired.

13. From a consideration of the location of the shafts with reference to the hull, the draft of the vessel and the lines of the after body, a figure is reached which places a maximum limit on the diameter of the propeller. The clearance be-

tween the tip of the blade and the hull should be kept large, because a small clearance interferes with the flow of water and lowers the efficiency. Next by use of the hull dimensions and Charts 17-A, 17-B, 20-B and 21-B as described in paragraph 6 and 7 above the Basic Slip and the factor K are determined.

14. At this point experience and study of similar installation are required to determine certain assumptions required to start the design of the actual propeller. On the cavitation chart Sheet 22-A the point of starting is picked within the following limits:

1. the value of $v \div V$ should not be greater than 1.0.

2. the value of $v \div V$ should not be less than the value given by the curve of the Basic Slip determined in paragraph (13).

3. there should be a margin of safety about 10 per cent above the Basic Slip line.

4. the e. h. p. ÷ E. H. P. is selected dependent on type of vessel approximately .30 for low speed vessels, about .70 for moderate speed vessels and .95 for high speed vessels.

With these general rules plus experience a point giving a value e. h. p. ÷ E. H. P. and for $v \div V$ is found. These are not final but may be used for a first approximation.

15. From Sheet 22-B using this e.h.p. ÷ E.H.P. and the Basic S, there is found the ratio of apparent slip ($s$) to Basic S, and from this the apparent slip is calculated.

Now, since $P = \frac{v \times 101.33}{R_d \times (1-s)}$ we can determine the Pitch.

16. There are various methods given in Admiral Dyson's book for proceeding from this point, but the most practical one is as follows:

With the maximum diameter and the pitch as determined assume a series of Projected Area Ratios and analyze each

for horsepower and revolutions. A curve then is constructed and the P*a* ÷ D*a* is selected which will give best efficiency, and still be out of the cavitation area. The pitch can then be varied to give desired R.P.M., as a small change in pitch does not seriously effect the S.H.P.

*Example:*

Assume from the hull lines value of K is 1.27

Assume also that the value of S is found to be .20

At the desired speed of 12 knots, the e.h.p. is 2000;

Approximate R.P.M. are 100. Propeller to be 3 bladed.

Maximum diameter = 19.0 feet

Type of ship — 2

Assume e.h.p. ÷ E.H.P. is .3

Then from Sheet 22-A for S = .20 above cavitation $\frac{v}{V} = .655$

from Sheet 22-B per cent S is .8

$\therefore s = .8 \times S = .16$

$$\therefore \text{Pitch} = \frac{v \times 101.33}{R_d \times (1 - s)} = \frac{12 \times 101.33}{100 \times .84} = 14.476 =$$

14.5 (approximate)

| Assume P*a* ÷ D*a* | .35 | .40 | .45 |
|---|---|---|---|
| (max) D | 19.0 | 19.0 | 19.0 |
| P | 14.5 | 14.5 | 14.5 |
| T.S. | 7600 | 8516 | 9450 |
| $S.T_D$ | 4.420 | 5.52 | 6.72 |
| S.H.P. | 9480 | 14100 | 19100 |
| $P.C_s$ | .707 | .675 | .638 |
| E.H.P. | 6700 | 9530 | 12180 |
| Basic S | .20 | .20 | .20 |
| 1—S | .80 | .80 | .80 |
| V | 14.56 | 16.36 | 18.12 |
| *v* | 12.0 | 12.0 | 12.0 |
| e.h.p. | 2000 | 2000 | 2000 |
| e.h.p. ÷ E.H.P. | .299 | .210 | .164 |

| | | | | |
|---|---|---|---|---|
| Curve (3) Sheet 21, $Z_p$ | .542 | .705 | .82 |
| log K | .104 | .104 | .104 |
| $Z_g$ | .438 | .601 | .716 |
| Curve (3) e.h.$p_g$ ÷ E.H.P. | .37 | .260 | .205 |
| $10^{Z_g}$ | 2.740 | 3.99 | 5.20 |
| S.H.P. | 3460 | 3540 | 3680 |
| $v$ ÷ V | .824 | .733 | .662 |
| Cavitating (Sheet 22-A) | No | No | No |
| Curve (1) Sheet 21 $Z_s$ | .15 | .305 | .435 |
| $10^{Z_s}$ | 1.41 | 2.017 | 2.725 |
| $S \times \frac{10^{Z_s}}{10^{Z_g}} = s$ | .103 | .101 | .113 |
| 1—$s$ | .897 | .899 | .887 |
| $R_d$ | 93·3 | 93·5 | 94·5 |

In examining this table we note the following:

*First* that the required shaft horsepower decreases as we decrease the P$a$ ÷ D$a$

*Second* that the screw with P$a$ ÷ D$a$ of .35 is the most efficient.

*Third* that the R.P.M. are practically constant, but lower than that specified: viz. 100.

*Fourth* that the $v$ ÷ V is increasing as P$a$ ÷ D$a$ is lowered. As stated above the maximum limit of $v$ ÷ V is 1.0 therefore it does not seem advisable to lower the area further and in order to leave a reasonable margin we settle on the first screw and proceed to vary the Pitch so as to get the desired R.P.M.

| | | | | |
|---|---|---|---|---|
| P$a$ ÷ D$a$ = | ·35 | ·35 | ·35 | ·35 |
| D = | 19.0 | | | |
| P = | 14.5 | 14.0 | 13.5 | 13.0 |
| T.S. = | 7600 | 7600 | 7600 | 7600 |
| $ST_D$ = | 4.420 | 4.42 | 4.42 | 4.42 |

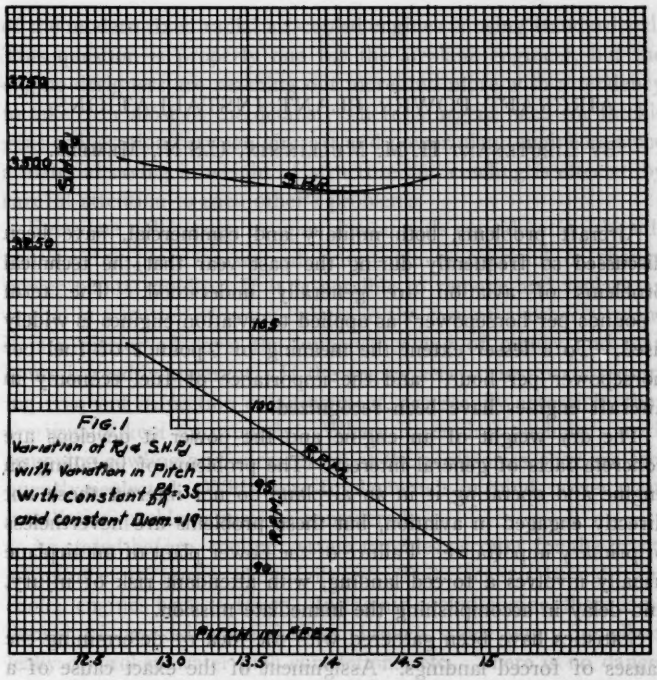
| | | | | |
|---|---|---|---|---|
| S.H.P. = | 9480 | 9150 | 8820 | 8500 |
| $PC_s$ = | .707 | .707 | .707 | .707 |
| E.H.P. = | 6700 | 6460 | 6240 | 6010 |
| S = | .20 | .20 | .20 | .20 |
| 1—S = | .80 | .80 | .80 | .80 |
| V = | 14.56 | 14.06 | 13.57 | 13.06 |
| e.h.p. = | 2000 | 2000 | 2000 | 2000 |
| $v$ = | 12.0 | 12.0 | 12.0 | 12.0 |
| e.h.p. ÷ E.H.P. = | .299 | .31 | .3205 | .333 |
| $Z_p$ = | .542 | .53 | .51 | .49 |
| log K = | .104 | .104 | .104 | .104 |
| $Z_g$ = | .438 | .426 | .406 | .386 |
| $e.h.p_g$ ÷ E.H.P. = | .37 | .39 | .405 | .425 |
| $10^{Z_g}$ = | 2.740 | 2.665 | 2.545 | 2.43 |
| $S.H.P_d$ = | 3460 | 3430 | 3465 | 3500 |
| $v$ ÷ V = | .824 | .853 | .883 | .919 |
| Cavitating | No | No | No | No |
| $Z_s$ = | .15 | .12 | .075 | .03 |
| $10^{Z_s}$ = | 1.41 | 1.32 | 1.187 | 1.07 |
| $s$ = | .103 | .099 | .0933 | .088 |
| 1—$s$ = | .897 | .901 | .9067 | .912 |
| $R_d$ = | 93.3 | 96.5 | 99.2 | 102.5 |

Construct curves Figure 1. based on pitch showing $R_d$ and S. H. P.; from these, we see that to get R. P. M. of 100 we should use a pitch 13.4, and we must install an engine capable of developing 3470 S. H. P.

The final propeller would have the following dimension:

| | |
|---|---|
| P$a$ ÷ D$a$ = | .35 |
| D = | 19.0 |
| P = | 13.4 |
| No of blades | 3 |
| Form of blade— | Standard. |

The above calculations illustrate the following points which every designer should have in mind:




(1) The efficiency of propellers increases with decrease in projected area ratio. But the limits are set by the necessity of keeping out of cavitation.

(2) The power does not vary greatly with change in pitch, but the R. P. M. increase with decrease in pitch.

(3) Use maximum diameter of screw, depending on hull structure and location of shafts.

In conclusion it should be remembered that the above discussion is elementary and that the various points are elaborated and fully discussed in Admiral Dyson's book. To perfect one's self in use of the tables and curves, the numerous examples in the book should be reworked and studied, so that the effect of variations in hull or propeller conditions are thoroughly understood. In the words of the Navy setting-up exercises "Continue the motion".

# AIRCRAFT POWER PLANT INSTALLATIONS.

By Lieutenant H. M. Mullinnix, U.S.N., Member.

Aircraft problems, both military and commercial, have been discussed so frequently during the past year that the technical problems of aviation are generally understood. The term "pounds per horsepower" as applied to aviation engines is widely used. To a lesser extent the meaning of "pounds of fuel per horsepower per hour," and the importance of fuel economy in aircraft engines have been recognized.

The dry weight of an engine and the power it develops are concrete facts of general interest. The problems of installing an engine and operating it in an airplane do not interest those not directly engaged in aviation, but these problems are nevertheless of primary importance. Failure of the power plant of an airplane usually involves a forced landing, with attendant risk of injury, and delay in accomplishing the immediate mission.

Statistics have been gathered with the idea of determining the causes of forced landings. Assignment of the exact cause of a forced landing is often impossible and the term "engine trouble" has proved very elastic in covering a multitude of errors of omission and commission. Inferences drawn from statistics must therefore take into account these uncertain factors. In general, it may be stated that power plant troubles cause about 85 per cent of the total number of forced landings. Of this percentage, aside from exhaustion of fuel in flight through ignorance or carelessness, cooling system troubles account for nearly half. Fuel supply and lubricating systems are responsible for as many more, ignition systems for a limited number, and the remaining insignificant number are actual mechanical failures of the engines proper.

The installation, that is, suspension of the engine in the structure of an airplane, is receiving more and more careful study. Mounting of water-cooled engines is more or less standardized and that

of the modern fixed radial air-cooled engines is being standardized. This question is properly one for the airplane engineer, but the engine designers are reaching out more and more with a tendency to take over most of the details. There is urgent need for intelligent coöperation between aircraft and power plant engineers since a material saving in weight of structure and cowling is effected by careful attention to each other's problems.

Everyone will recognize upon a moment's consideration the advantage for peace or war of being able to overhaul or change power plants rapidly. If it is possible to remove and replace an engine in a matter of hours, where in the case of another airplane it is a matter of days, the immense saving of money and man hours is readily apparent. If the engine accessories, parts requiring frequent inspection or adjustment, are readily accessible, they are usually kept up properly. Where it requires an hour's work removing cowling and wires to get at the accessories, they are frequently neglected. With these fundamentals in mind, power plant installation problems may be considered.

As stated before, cooling system troubles account for the greatest number of power plant failures. This fact has brought air-cooled engines back to favor. In the air-cooled engine, there is no danger of cracked water jackets, broken water pump impellers, leaky or broken water pipe connections, etc. There is no danger of boiling away cooling water when temperatures are high, and there is no danger of frozen radiators when they are low.

Since there are now in use many more water-cooled engines than air-cooled engines, and since the use of these will continue for many years, attempts are being made to eliminate the above difficulties. Radiator construction has been carefully investigated and improved. Engine construction has been developed to the point where water leaks from the jackets themselves are relatively rare. Anti-freeze solutions have been tested and proved. One product in particular may be employed to prevent freezing at extremely low temperatures, and possesses, in addition, the very desirable characteristic of boiling at a temperature much above that of water. In line with current activity in the automotive industry, the possibility of utilizing steam cooling for aircraft engines is being investigated with considerable prospect of ultimate success.

Air-cooled engines, although free from the multitude of ills affecting water-cooled engines, have some difficulties of their own. Adequate means of shuttering the air-cooled engine cylinders involves intricate mechanisms and the apparatus has been too complicated for general use. It is not possible as in the case of the water-cooled engine to facilitate starting in cold weather by filling the jackets with boiling water, neither is there any convenient and simple means for supplying heat to the oil piping to speed up attaining the normal operating oil pressure. These difficulties will require hard study and intensive development, but a solution for each is in sight.

Under cooling, it is worth while mentioning the question of oil temperature regulating apparatus. Lubricating oils employed in aircraft engines congeal at moderate temperatures, having pour points around the freezing point of water. When an engine is started in cold weather, the maximum suction the oil pressure pump can exert is not sufficient to overcome the viscosity of the cold oil in the suction pipe. There is grave danger of damaging the cylinders and pistons before the engine and its connections have attained a normal operating temperature. On the other hand, there is so much heat rejected to the lubricating oil by most engines under full throttle operating conditions that outside means for reducing the temperature to a safe point must be provided. For water-cooled engines, an oil regulator has been developed. This unit permits interchange of heat between the lubricating oil and the cooling water. During the warming up period of the engine, the radiator is bypassed and the water circulating rapidly through the jackets and regulator heats up and warms the lubricating oil. Then when the radiator is cut into the water circulating system the cool water passing through the oil regulator brings the oil temperature down to a safe value.

Next to the cooling system, the most troublesome problem in aircraft engine operation is that of supplying the engine with fuel. At first glance, this seems relatively simple. In modern automobiles, for instance, there are three systems in general use. First, straight gravity feed; second, vacuum tank supply system, in which fuel is drawn by intake manifold vacuum into a small gravity tank, and, third, air pressure supply.

The vacuum tank apparatus is covered by patents and every unit bears a patent royalty charge. Inventive genius has been concentrated for years on finding a satisfactory substitute, but the vacuum tanks are still in general use.

In aircraft, vacuum systems are not possible, because aircraft engines operate usually with relatively small intake manifold depressions, and the influence of changes of atmospheric pressure with altitude is too great. Air pressure systems have been extensively used in aircraft and provide one of the surest and simplest fuel supply systems. They have fallen in disfavor chiefly because the presence of even a slight air pressure in a large fuel tank has proved a fire menace in case of a crash. In addition, to withstand the air pressure involves additional stiffening of the tank sides and therefore extra weight.

Gravity systems are employed wherever conditions permit, and for training planes gravity fuel systems are preferred. For airplanes designed for long distance flights, especially with high power engines, it is not practicable to carry any large proportion of the fuel supply sufficiently far above the carburetors to insure adequate flow of fuel throughout the normal range of flying attitudes. Where the gravity fuel tanks are located well back of the carburetors a change in attitude of the airplane through a range of some 15 degrees may reduce the head of fuel on the carburetors to such a degree as to stop the necessary flow. Gravity systems are therefore of limited application in military aircraft.

The approved type of modern fuel systems employs a positive fuel pressure supplied by some type of pump. Wind-driven pumps, that is, pumps operated by small propellers or wind mills, have been extensively used and still remain one of the simplest and most reliable forms of fuel pump installation. Wind mill pumps may be located in various positions to suit the installation and particularly may be placed near the bottom of the fuel tanks, where the pump will be effectively drowned and therefore always primed.

Wind mill pumps are inefficient in power consumption, especially for high speed aircraft. They are subject to damage from carelessness of unskilled personnel or from spray or loose debris thrown up by the propeller slipstream. Pumps operated directly from the engine are preferred.

Various types of fuel pumps have been tested extensively, but mention need be made of three only. Sylphon or bellows pumps are very efficient and operate satisfactorily against a high suction lift. They are used successfully in racing automobiles and have been widely used in aircraft. As originally installed with a positive connection to some reciprocating engine part, the bellows proved unreliable. Sylphon pumps were discarded for that reason, although, in a modified form, where operated by a cam with a spring return and amplitude of reciprocation proportional to the demand for fuel, this defect is overcome.

Plunger pumps have been employed, but present great difficulties in packing and lubricating the plungers. The most successful type employs an oil pressure seal, balancing lubricating oil pressure against fuel pressure. This type has many advantages, but is subject to one serious disadvantage, the danger of passing fuel into the lubricating oil while the engine is not in operation and there is no lubricating oil pressure.

Gear pumps, are small, light, simple, and reliable. They cannot be operated, however, against any considerable suction head of fuel and must be kept down near the level of the fuel. In using a gear pump, it is customary to locate it at the lowest possible point on the engine. If the engine is mounted too high, it is possible to drive the pump through a flexible shaft. Since the necessary power is slight, this latter arrangement permits locating the gear type fuel pump advantageously without appreciable loss of reliability.

No fuel system is entirely proof against failure. In the interests of reliability, it is necessary to duplicate the means for supplying fuel to aircraft engines. There are two common methods of achieving this; first, by providing a small gravity tank containing a reserve supply of fuel; second, by providing a hand-operated fuel pump paralleling the engine-driven fuel pump. Both methods are effective.

The question of lubricating oil regulators was touched upon in the discussion of cooling systems. Practically all modern aircraft engines operate on the dry sump plan. Excess oil thrown off the bearings drops into the crankcase and is evacuated by means of two or more scavenger pumps arranged to draw oil from opposite

ends of the crankcase. This arrangement takes care of all reasonable changes in inclination of the engine within the normal range of flying attitudes.

Where the engine power is low or the rejection of heat to the lubricating oil is small, the scavenger pumps discharge directly to a supply tank from which the oil pressure pump draws the oil which it forces through the engine. Some of the excess heat picked up by the oil is dissipated through the tank walls and any foam discharged with the oil from the scavenger pumps rises to the top of the tank, where adequate expansion space is provided. In typical oil systems such as outlined above, the "plumbing" is the chief source of difficulty. Practically all troubles are due to breakage of pipes from vibration, or loosening of pipe connections. There are isolated cases only of leaky tanks or plugging of an internal oil passage in the engine itself. Strainers are provided, of course, to catch particles of foreign matter large enough to injure fuel pump gears or plug up passages.

Where the heat rejection to the lubricating oil is high, it is necessary to provide additional cooling for the oil. Various plate or core type radiators are used, each having some particular drawback. When the cooler is located on the discharge side of the scavenger pumps, the oil is hot and foaming. Presence of entrained air in the foam reduces the efficiency of heat transfer, making the cooler unit inefficient. The efficiency of the cooler is further reduced by the high viscosity of the film of cold oil next to the cooling surface. If the cooler is located on the suction side of the pressure pump, the trouble with foam is avoided, but a material resistance is added in the suction line. With cold oil, such as is encountered in starting an engine in cold weather, this added resistance is sufficient to prevent attaining satisfactory oil flow in a reasonable time.

The oil regulator mentioned before is a partial solution for the above difficulties. In its present form, it has proved a material aid to operation of water-cooled engines under a wide range of temperature conditions. A serious objection to this unit is that its use introduces an additional danger of loss or contamination of lubricating oil. The best solution seems to be the use of an air cooler. This should be mounted on the suction side of the pressure

pump, with a manually-operated bypass for use while starting the engine and provision for employing exhaust heat to warm the oil in cold weather. Another possibility is to reduce the quantity of heat rejected to the lubricating oil by changes in the engine lubrication and improvements in the direct cooling of the cylinder heads and pistons.

The ignition systems of aircraft engines in current use are confined to two types, battery generator such as employed by ordinary automobile engines, and magneto systems. It is the universal custom to duplicate these systems, using either two entirely separate high tension distributors or two magnetos. Advocates of the battery generator systems claim various advantages. The system is flexible and comparatively easy to lay out and install. It gives a "hot" spark at the extremely low crank speeds used in starting aircraft engines. It is frequently necessary to employ an engine-driven generator to charge batteries for lighting or other purposes, and the same generator can be used to charge an ignition battery.

In the face of those advantages, the employment of magnetos is increasing. There is not much to choose in weight between dual magnetos and a battery generator system complete with battery. The magnetos, however, are assembled with the engine as a complete power plant unit. There is no trouble from worn-out or run-down batteries. The magnetos do not deteriorate in storage or while temporarily out of service. In airplanes with radio equipment, magneto ignition simplifies the problem of ignition shielding to prevent interference with radio reception. In case of a crash, the magneto ceases to function when the engine stops and there is no fire hazard such as results from the presence of a live battery which may be short-circuited by debris. Modern aircraft magnetos are so exceptionally reliable that their average life exceeds that of the engines employing them. These magnetos require a minimum of maintenance attention and greatly reduce the danger of ignition system failure resulting from the carelessness or incompetence of mechanics.

There are an infinite number of details in connection with aircraft power plant installations which could be discussed. Each type of aircraft presents problems peculiar to itself, but it is not possible to go into these details. No mention has been made of

the problems and possibilities of solid fuel injection, of supercharging, or similar endeavors.

In conclusion, there are four systems essential to the operation of the common aircraft engine. These are fuel, ignition, lubrication, and cooling. The troubles with aircraft power plants are confined almost entirely to failure of one or more of those systems. "Plumbing" is the chief offender with all other sources of trouble combined a poor second. Experience and ceaseless development, coupled with improved materials and methods of manufacture, have brought the modern aircraft engine to the point where an actual primary mechanical failure is almost unknown.

In spite of their extremely high power output and low weight, modern aircraft engines are very reliable. There are authentic records of engines in Naval aircraft which have operated approximately 300 hours' flying time without major overhaul. Three hundred hours represents nearly 30,000 miles, or five years' mileage for the average motor car. In contrast with the usual operating conditions of an automobile engine, aircraft engines must function normally at or above 75 per cent of their maximum power. Engines representing modern types have successfully completed full throttle endurance tests ranging from fifty to several hundred hours' duration. These engines represent the highest refinements in designing skill in a mechanical age.

Power plant failures still occur in the operation of aircraft. A high duty aircraft engine cannot be made entirely fool-proof. Designers are bending every effort to improve their products. The operating personnel must be equally keen to improve operating conditions. With reasonable skill and attention on the part of operating personnel, modern aircraft power plants may be depended upon for efficient, reliable and dependable service.

# ABSTRACTS FROM THE GERMAN PRESS.

By E. C. Magdeburger, Member.

## 27th Annual Meeting of the German Shipbuilding Society, 18-20 November, 1926. Schiffban 1 December, W R H 7 Dec., 1926.

Most important among the papers presented before this meeting were the two papers devoted to efforts to improve the efficiency of steam on shipboard, (1) The Modern Steam Turbine and Its Use for Ship Propulsion—by E. A. Kraft, and (2) Possible Developments in High Pressure Steam for Marine Application—by O. H. Hartmann.

### (1) The Modern Steam Turbine and Its Use for Ship Propulsion, by E. A. Kraft.

With the use of geared drive on shipboard the marine turbine became independent of propeller speed and could appropriate the wealth of experience gained with steam turbines ashore. The author divides the means available for the improvement in efficiency in three groups. (*a*) The available heat can be increased by increasing the pressure and temperature of the fresh steam on one side and by lowering the exhaust pressure on the other. The latter, however, is limited by the temperature of the cooling water. Figure 1 shows that the theoretical thermal efficiency of the cycle increases first quickly then much slower. This being one of the reasons why in condensing turbines initial steam pressure is generally not higher than 35 atmospheres or 425 pounds per square inch. Another reason is, that the moisture in the exhaust shown in the lower part of Figure 1 increases materially. The second group of means to improve turbine efficiency are improvements in the cycle used which became very important when the

question of high pressures was definitely attacked. To this group belong the preheating of the feed water by means of interstage bleeding and intermediate superheating of the working steam. The greatest progress in the steam turbine industry of recent years was made in the improvement of the efficiency of the

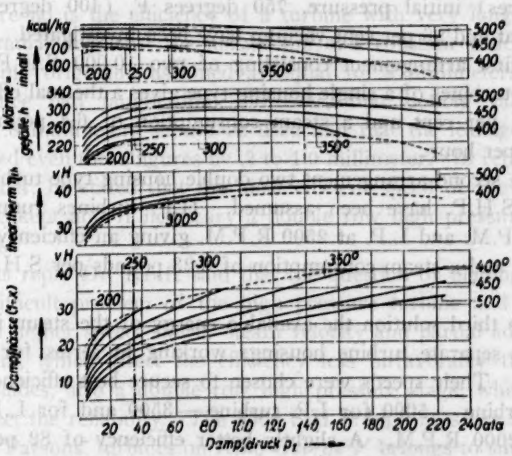



FIGURE 1.—ADIABATIC EXPANSION OF STEAM OF DIFFERENT INITIAL PRESSURE DOWN TO .6 PER CENT VACUUM. ATMOSPHERES = 14.2 POUNDS PER SQUARE INCH.

Dampfdruck — steam pressure.
Dampfnässe — Steam moisture.
Theor. therm. ηth—theoretical thermal efficiency.
Wärme — heat; gefälle h — available heat drop.
Inhalti — contenti; Kcal/kg = 1.803 B.t.u./lb.

turbine itself. It is realized now that not the slow steam velocities in themselves, but mostly their consequences, such as, increase in the ratio of working steam area to the area that must be kept tight, are the cause of the improved efficiency. A further requirement of efficient blading is that the increase in working area and blading length occur gradually to reduce eddy currents in the dead spaces.

Further means of improving the efficiency is the steady increase in the power developed per unit and especially in the limit developed in one housing at any given speed. A modern 3000 R.P.M. turbine may be built now up to 40,000 H.P. As an example of attainable efficiencies, four different solutions for a twin screw ship of 20,000 total S.H.P., assuming 425 pounds (35 atmospheres) initial pressure, 750 degrees F. (400 degrees C.) superheat and 96 per cent vacuum have been investigated.

The first arrangement consisting of two 10,000 S.H.P. 3000 R.P.M. turbines of a single housing type gives a thermal efficiency of 78½ per cent and a steam consumption of 6.4 pounds per S.H.P. per hour.

In the second arrangement two double housing type turbines of 10,000 S.H.P. have been assumed. H.P. turbines running at 5000 R.P.M. and L.P. at 2500 R.P.M. giving an efficiency of 81 per cent and a steam consumption of 6.23 pounds per S.H.P. per hour.

In the third solution the available energy of the steam is used in three separate turbine housings working in series for ahead running. Their speeds were chosen to secure best efficiency, for H.P. turbine — 5000 for I.P. turbine — 3500 and for L.P. turbine — 2000 R.P.M. A slightly better efficiency of 82 per cent could be obtained with steam consumption of 6.14 pounds per S.H.P. per hour.

In the fourth solution there are two turbines, one per each shaft, working in series. The H.P. unit with 3300 R.P.M. and the L.P. turbine with 2000 R.P.M. An efficiency of 83 per cent can be obtained and the steam consumption lowered to 6.05 pounds per S.H.P. per hour. The above efficiencies refer to pure condensing operation, a reasonable number of stages and substantial blade clearances, such as can only be considered for Marine purposes.

In the following discussion Dr. G. Bauer asserted that the time appeared to be ripe to energetically push the introduction of high pressure steam for marine propulsion. Since the efficiency of the turbine as stated by the author was unfavorably influenced by the short blade length, he wished to know what means there are available for overcoming that deficiency.

Prof. G. Flugel from his experience in the turbine department of the A.E.G. emphasized the influence of blading length on efficiency, and stated that $\eta$, the efficiency of the turbines with short blades can be expressed by the formula $\eta = \eta o \left(1 - \frac{\alpha}{L}\right)$.

Where $\eta o$ is the efficiency of a turbine with very long blades, L the radial height of the nozzle and $\alpha$, a constant which latter was being originally assumed as equal to .8 millimeters. The experience had taught him, however, that double values or 1.2 — 1.8 millimeters should be used and that the leakage losses increased even these figures by .2 to 1.0 millimeter. In replying to Dr. Bauer, he stated that the most efficient speed of the turbine must be kept and if necessary a double row high pressure stage be used.

In his reply Dr. Kraft said that the question of blading is the most difficult problem of the high pressure turbine and recommended higher speed, slower steam velocity and partial admission of steam, which affect the efficiency less unfavorably than too short blades, also a double row high pressure stage, which does not affect the reliability. The 3500 S.H.P. high pressure installation of Parsons' turbines on *King George V* belongs to small type and is evidently fitted with very small clearances.

(2) Possible Developments in High Pressure Steam for Marine Application—by O. H. Hartmann.

The author has on numerous previous occasions reported on the progress of the investigations made by his firm on the subject of high pressure steam, which subject has of late become so popular with the steam engineer. He pointed out this time the importance of the experimental installation on the steamship *King George V*, consisting of Yarrow boilers delivering steam of 600 pounds per square inch and Parsons' turbines, and wanted to see in it a courageous bid by English engineers to regain the leadership lost with the introduction of superheated steam and oil engines.

The following table shows the economies possible with the higher pressure steam:

| | | | | |
|---|---|---|---|---|
| Absolute initial pressure in pounds per square inch | 225 | 540 | 900 | 1500 |
| Heat Consumption in B.T.U. per S.H.P. | 8950 | 8000 | 7260 | 6740 |
| Saving possible in per cent | ..... | 9.9 | 18.85 | 24.6 |

Assuming 85 per cent for boiler efficiency, a heat consumption of 6740 B.T.U. per S.H.P. corresponds to .59 pound of coal, and .44 pound of oil which is not very far from the best consumption obtainable with Diesel engines. Since boiler oil is generally cheaper than the oil used in Diesel engines, a high pressure steam installation has already an economic advantage over the Diesel engine, provided, of course, that the first cost of the two is comparable and both are equally reliable. Hence designs must be developed that will assure to high pressure steam such reliability on board and it is wiser to gather experience on smaller craft such as towboats and small freight ships.

The number of high pressure boilers is constantly growing and some of them have been in operation for several years already. These boilers are of the water tube type with seamless drawn drums, as a rule. Fears that high temperature steam and water may attack the walls have been proven groundless. Schmidt Company's* boiler has shown no ill effects from that despite the 18,000 working hours with water from 480 to 570 degrees F. and superheated steam of 880 degrees F. Nevertheless the water tube boiler is not very popular with marine engineers.

The author then devoted some time to the description of boiler types that do not have the disadvantages of the water tube boiler and which in his opinion may be suitable for marine installations. Among these were so-called double pressure boilers in which, to reduce the first cost, only the high temperature gases are used for steam making and the feed water preheated as high as possible by means of separate less costly heating surface arranged farther along the path of the gases.

The boiler producing working steam in such designs is not heated by direct contact with products of combustion, but by steam of 200-300 pounds per square inch higher pressure, and this distilled water used for heating circulates continuously without having to be renewed. Such a boiler (Figure 2) is explosion

* The author is connected with "Schmidt'sche Heissdampf G.m.b.H.," Cassel-Wilhelmshöke.

proof, since any possible damage to a tube will result in liberation of a small amount of heating water.

However, high pressure boilers with but one working pressure are also claimed to be quite reliable and they permit considerable reduction in weight. Experiments on such a boiler have demonstrated that up to 4700 pounds of steam could be generated per square foot of heating surface.

As important as the boiler is for generation of high pressure steam the steam engine is for proper utilization of it. For small installations reciprocating engines, expanding the steam in 5 stages arranged on four cranks have been advocated. Simple superheaters between stages contribute much to the ultimate economy of the installation. Piston valves have proven reliable under highest pressures and temperatures. Combined engines and turbines as proposed by Bauer-Wach (shown elsewhere in this issue) are also well adapted, when proper efficiency in high pressure turbines has been secured these also may be used. However, until successful high pressure turbines have been developed reciprocating engines must be relied upon at least for the high pressure stage.

The author recommended that shipping men find sufficient courage to use pressures not less than 750-900 pounds per square inch and not to waste time on developing intermediate pressures of 450-600 pounds. Boilers and piping today do not offer any more difficulties to the higher pressures than they do to intermediate range of pressures.

In the following discussion Dr. Wach pointed out that lowering of the expansion pressure with reciprocating engine results in excessive dimensions of the low pressure cylinder. The addition of a low pressure turbine utilizing exhaust steam from the reciprocating engine resulted in 30 per cent increase of power on the steamer *Sirius*—450 S.H.P. without and 600 B.H.P. with turbine. Similar results have also been secured on a larger installation on the steamship *Elberfeld* where the turbine added 730 S.H.P. to the 2350 S.H.P. of the engine and it required only 3 weeks to install it.

Director Goos of the Hamburg America Line was inclined to think that Wm. Denny was right in selecting 600 pounds steam pressure and 750 degrees F. temperature for the *King George V*

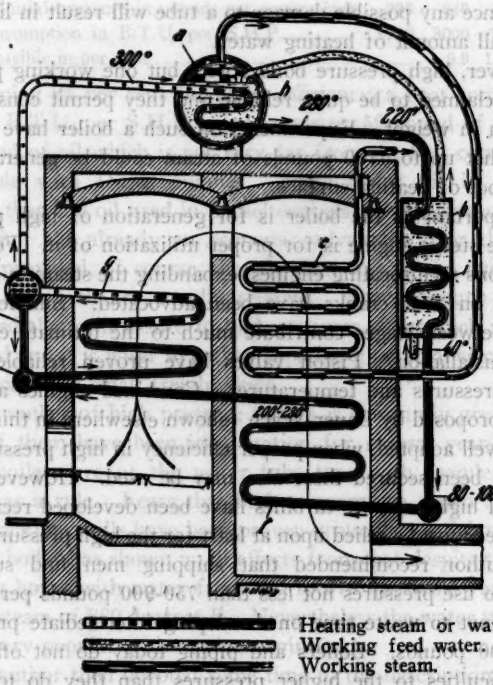



FIGURE 2.—SCHEMATIC ARRANGEMENT OF BOILER FOR HIGHEST PRESSURES WITH INDIRECT HEATING PROPOSED BY SCHMIDT'SCHE HEISSDAMPF G. M. B. H.

a—Working steam boiler.
b—Feed water heater.
c—Working steam superheater.
d—Water separator for heating steam.
e—Water header.
f—Preheater.
g—Steaming coil for heating steam.
h—Heating coil.
i—Heat exchange coil.
k—Equalizer pipes.

installation and the proposal of the author to go right away to higher pressures appeared to him too risky, since previous experience on board ship with steam of 840 degrees F. was not encouraging. Feed water and condensers appear to be the problems. The smallest amount of salt makes water tube boilers prime so intensely that their load must be reduced. Hence instruments to indicate accurately salt contents in the feed water and means for isolating the salted parts must be provided. Besides, authoritative opinion differs on the efficiency and fuel consumption figures actually attainable. The shipping men ought to create means for finding out the facts.

In replying the author pointed out that Director Goos was hardly right in his objections, since one of the main advantages of the high pressure boiler was the fact that a small quantity of salt has no effect on it and even scale does not form in this type of boiler, as his firm's experience has shown. He also mentioned that Yarrow type of boiler is not very suitable for the purpose and in his opinion the problem will not be solved by this type of boiler.

BAUER-WACH COMBINED RECIPROCATING ENGINE AND EXHAUST STEAM TURBINE. WERFT-REEDERI-HAFEN. 7 OCTOBER AND 22 NOVEMBER, 1926.

In order to improve the efficiency of steam engines used in ship propulsion by better utilization of available heat, a steam turbine must be added, using the exhaust from the reciprocating engine. The torque developed by this turbine must be transmitted to the engine shaft by means of a gear drive with a hydraulic coupling of the well-known Föttinger type between the two. This is essentially the solution arrived at by two eminent German Marine engineers, Wach, director of the Tecklenborg Shipyard, and Prof. Bauer of Vulcan Works, and now used by these two shipbuilding organizations.

The principal advantage of such a solution lies in the fact that the transmission of the torques developed by the slow running steam engine and the high speed steam turbine to separate shafts is economically unattractive.

The exhaust steam turbine is used only for ahead running. When running astern the turbine is disconnected by emptying the hydraulic clutch, so that only the gear and the comparatively slow running first pinion rotate with the engine shaft in reversed direction. An oil distributing valve controls the filling of the clutch and the consequent coupling of the steam turbine when again running ahead and only when this coupling is accomplished can exhaust steam enter the turbine.

When turbine is out of order the gear drive can be disconnected. On the other hand when the reciprocating engine is out of commission fresh steam can be admitted direct into the turbine and 40 per cent of the original torque supplied to the shaft.

The gear drive is so arranged that the wear of main bearings of the reciprocating engine has no influence on the accurate meshing of the gears.

The first set of this type was installed on the high sea fishing steamer *Sirius* and is shown on Figure 3 while undergoing shop tests. A fuel economy of 25 to 30 per cent as compared with an installation without the added steam turbine is claimed to have been obtained. The simplicity of such additional power to obtain more speed for an existing installation or reduce the engine room space for a new installation has won the attention of the German shipowners and two more ships—*Elberfeld* 9350 tons of North German Lloyd and *Frauenfels* 11,355 tons of the *Hansa* Line have been contracted for by the Tecklenborg yard. The former has run successful trials on 6 November, 1926.

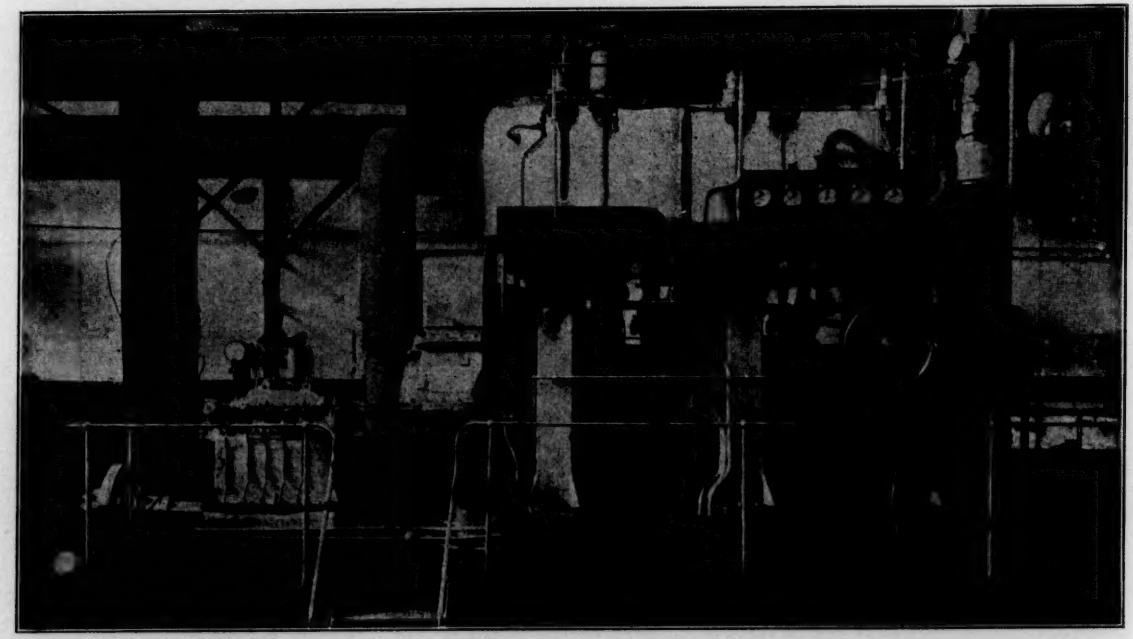


FIGURE 3.—COMBINED STEAM ENGINE AND EXHAUST STEAM TURBINE ON THE TEST STAND.

# NOTES.

## ENGINEERING.

By Charles Edward Lucke, New York, N. Y.

Engineering is a profession and as such stands beside law and medicine. It is not a trade, though it is in contact with the trades. It is not a business, even though it has become a necessity to modern big businesses. Nor is it a science, though based on the exact sciences of physics, chemistry, and mathematics, and in its practice the engineering method is essentially scientific.

Engineering is concerned with utility, with the creation of services and things for public use, especially new kinds, better, more useful or more economical than were available before. All of the discoveries of scientific study and research are sought by the engineer as a possible basis for some new sort of product, some improved process, or some better public service. The development stage, from the conception, the way in which some betterment might be effected, to its commercial realization, and, in addition, the most economical handling of the new thing, is strictly the engineer's affair. In this he acts as a director of the enterprise or technical expert over the whole or over any one of the more important or complicated parts. It is here that engineering becomes a bond between the activities of other groups, including all of the trades, the labor and skill of which do the actual work planned by the engineer; the bankers, who must find the funds to finance the enterprise; the industrial leaders, whose administration is needed; the merchants, who link producer with consumer; and at every step the lawyers, or, in the case of problems of more strictly public nature, the appropriate governmental departments or bureaus.

With professional activities so widespread as this, and so intimately knitted into the fabric of affairs of all the advanced nations, it is rather strange that so small a part of the public does really understand what engineering means, or what constitutes an engineer. Many explanations for this situation have been advanced, but of them all only two need be mentioned. The first is that the engineer acting in his professional capacity is usually an agent, not a principal; he is working out something or directing some undertaking for some one else, an individual, a corporation, or the Government, and the public hears of the principal but not of the agent. The second explanation is found in the attitude of the engineer himself, who, dealing with *things* primarily, the materials and the forces of nature, is so absorbed in the great natural laws, which he is trying to utilize in some new or better way, that he has developed a sort of impersonal attitude and an unwillingness to talk. The *thing* is so big and so absorbing that mere words seem trivial and inadequate, and talk a waste of valuable time.

Every discovery in the basic sciences, or in the principles of engineering, every invention and every engineering development gives birth to a new family of discoveries, inventions, principles, methods, and products, and

this process results in a multiplication of ideas and useful results every year. As a consequence, engineering is growing in scope at an ever-accelerating rate in a sort of geometrical progression, somewhat like the multiple compounding of interest on investments. This fact has forced engineers, in the practice of their profession, to limit their activities to some one division in which it is possible for one man to become really expert, an authority; and so engineering itself has divided into definite branches, each well recognized. This division is still going on, and there is every reason to believe that further division will be made in the future, and new branches of the profession will be recognized as legitimate.

At the present time there are four major divisions of engineering that are fully recognized, and many others only partly accepted as proper divisions, and so of minor importance, either because of smallness of number of members or because of lack of real justification.

The four major divisions include mining and metallurgical engineering, civil engineering, mechanical engineering, and electrical engineering.

*The Mining Engineer* concerns himself with the location of mineral deposits, including all the fuels and the ores of the metals, but it is his special problem to find and to apply the most economical methods of getting these natural resources out of the ground and into the general market for use. In the case of some of these materials he also undertakes the development of methods of treatment to get from them the greatest values or most useful materials derivable from them; and in the case of metals the metallurgical engineer directly cooperates by receiving the ores, removing the crude metal, and refining and treating each part up to the point of producing metals of all required properties in the most useful standard shapes and sizes.

*The Civil Engineer* plans and directs the construction of the great public works; highways and railroads; bridges and tunnels; river and harbor improvements; canals; water supply and sewage systems, including big-scale sanitation and land-reclamation projects, and other constructions of related nature.

*The Mechanical Engineer* deals with machinery of all kinds and with related tools, appliances, implements and apparatus, mainly constructed of metal. This includes machinery for generating power from any of the fuels, or from the water of streams; machinery utilizing power to manufacture some commodity, the so-called process machinery to produce textiles, cigarettes, printing, metal and wooden goods; machinery for some special service, such as pumping and refrigeration, heating and ventilating; machinery and apparatus for the other divisions of engineering to use, such as mine hoists, blowing engines for ore smelters, dredges and excavators, and equipment for chemical processes. It also includes that special large and important class, the machinery of transportation, including the driven or carrying vehicles, steam and motorships and motorboats, locomotives and rail cars, automobiles, trucks and tractors, and aircraft; and, in addition, every sort of equipment for fuel utilization for any purpose whatever.

*The Electrical Engineer*, as the name implies, is skilled in all that concerns electricity and magnetism. He assumes responsibility for the utilization of electrical and magnetic principles, the development of equipment to get the most practical and economical results, and also, in the many cases where this is complicated, he undertakes its most efficient handling when it is put to work. This equipment includes electric generators for converting the mechanical power generated by engines and turbines into electricity; electric motors for converting electricity back again into mechanical power; and also the transmission of electric power from any number of generators to distant motors in any number. This electric transmission includes that

large-scale enterprise of connecting many central power-generating stations with each other and with widespread transmission systems, even extending across several states, now known as superpower. In addition, there must be noted innumerable electrical appliances utilizing electricity as such, as distinguished from its utilization for power. Into this group fall the electric heating appliances, such as the large furnaces of the electro-chemical industry, and also the electric lamp, the basis of all modern illumination. Finally, there is the particularly important and perhaps best-known class of apparatus in which electricity serves as the key to communication, that of the telegraph, telephone, and radio systems, which, acting with transportation, accomplish the linking of distant places, bringing their people and their products together.

The engineering profession is a highly organized one, and the evidence of this, independent of its actual practice day by day, is found in the engineering societies, and in a most voluminous and fast-growing body of literature in the form of books and periodicals. Each of the four major divisions of engineering is represented by a national society in this country, with an aggregate membership of over fifty-five thousand, cooperation being secured through a central organization, which is in effect the national representative of the engineering profession. There are equivalent societies abroad, all maintaining contact and working together for the advance of engineering, without regard to language or national boundaries. In fact, engineering has developed a sort of international language of its own—its terms, its drawings, its methods, its materials, its appliances, and its products are understood by the engineers of all nations, regardless of their spoken language.

In addition to the purely professional societies of the four major divisions noted, and perhaps a dozen or so minor ones, such as the chemical, marine, railroad, heating and ventilating, refrigeration, and automobile groups, there are perhaps a hundred semi-professional associations in which engineers join with business men, with bankers, or with men of the skilled trades in promoting some common interest involving problems beyond the scope of any one class. Among these may be mentioned, for purposes of illustration the National Electric Light Association, which is the central organization representing the business of generating electricity in central electric power stations and of its distribution to consumers, with a membership of fifteen thousand. The American Foundrymen and the Railroad Master Mechanics are also illustrations of this sort of associations of semi-professional, semi-business character. While no statistics are available, the aggregate membership must be somewhere near a quarter of a million in the United States alone.

To enter the engineering profession a man must be trained, and a fair percentage must, in addition, be well educated to insure the vision and sympathetic contact with other men, especially non-engineers, that is so necessary to leadership of even the best-trained professional man. This training and education of engineers is also highly organized both in this country and abroad, and the very large number of students graduated each year by the engineering schools is immediately absorbed, with at least an equal number of men who succeed in rising from the ranks of the trades, and who, by study, qualify as professional engineers. In the engineering schools of the United States there are fifty-six thousand students enrolled, and eight thousand five hundred are graduated each year to begin the profession of engineering.

As a rule, the engineering schools of Europe are independent of the old universities, and in this country there are many such independent schools.

It is quite typical of the United States, however, that most of our best universities have established engineering schools on a par with the law school, the medical school, and with the non-professional departments of university scholarship. This is regarded as a great step forward, because of the resulting contact between the representatives of the different professions affecting professional standards, and, in addition, the contact of the professional-school professors and students with the great body of non-professional scholars making up the rest of the university.

In the practice of engineering probably the most characteristic element is the point of view of the professional engineer, the basis of the engineering method. This, while founded on the exact sciences of physics and chemistry, and, therefore, properly named scientific, does not stop here, but must, and actually does, include an economic factor and, in some cases, also a sociologic one. Briefly, the engineer considers himself always as having a problem to solve, a problem of utility of some sort. A way must be found to do something or get something done, and the best possible way. Perhaps that thing has never been done at all, or perhaps it has been done but not in a satisfactory way. The first stage in the solution of such a problem is to assemble all possible ways in which the thing might be done, drawing on history, on past practices, on established principles, and on recent discoveries. Extensive knowledge of the subject is essential here, but it alone is not sufficient, because there must be capacity to recognize what things might apply to the case, somewhat similar to the capacity of a judge in a court of law to distinguish relevant from irrelevant evidence, and to apply the proper law to the case. This is the period of exploration, and in it lies the same lure that forms the foundation and motive for any exploration. It calls for the courage and fortitude of the crusader who will not give up, and it brings the same reward—satisfaction.

It is quite the common thing for the engineer to discover that some of his possible solutions are new and patentable inventions, and it is in just this way that most of the really useful inventions are made. They are solutions of problems worked out in a systematic effort to do something that is needed. Incidentally, there are more inventions of this sort produced per year in the United States than anywhere else in the world, and this is direct evidence of the great activity and productive fertility of American engineers.

Of all the possible solutions of a problem, some will be more suitable in one respect and less so in others. Some will be best adapted to existing tools, to the skill of available labor or to most available materials, to such plant or manufacturing facilities as are at hand; some will be more efficient or have greater reliability than others, and some will be most pleasing to one or another class of users for whom the problem is being solved, whether they know it or not. These are all questions of relative suitability as measures of value of the possible solutions, quite independent of the other phase of economics, that of dollars and cents, costs and selling prices, as factors in value. Of the several possible solutions of the problem some will be adapted to cheaper raw materials, or be producible at lower production costs; some will require less investment in equipment or in stock than others, and, with the best combination of these factors, some of the solutions will be cheaper than others. In these studies of comparative economics in relation to possible solutions of a given problem, it is very rare indeed to find that any one solution is best in every respect, so rare as to be regarded as almost a miracle. Furthermore, it is almost equally unusual to find that the solution which is really best, as to qualities or elements of suitability for the

intended purpose, is also the cheapest; more often it is the most costly. Then comes the real judicial responsibility of the engineer because a new problem has arisen. Out of the two numerous solutions of the original problem of ways and means has now come the problem of economic selection, which requires an engineering decision. Engineers must make decisions, and in the long run they are themselves judged by their decisions, as is the case with other professional men, the lawyers and the physicians. All must be right much more often than wrong, and here the public is the final or real judge.

For a problem of major magnitude all this may take several years, as every doubtful point must be checked up and proved, even if costly experimentation is necessary, first in a laboratory and then later on a larger scale than is possible in a laboratory, but in the end there is always a clear conclusion, a real decision, and in it there is very little of the element of mere opinion, and no guessing. After the decision as to which solution of the problem will be the best, there comes actual commercialization under engineering direction, and finally the public receives a new product, or a new kind of service, or a better service of an old kind, without any knowledge of how it happened, and with no reference to the names of the engineers responsible for its creation. Thousands of such projects are under way all the time, and this will continue forever.

Public interest in and knowledge of engineering, while yet far less than in the case of law and medicine, is nevertheless on the increase, and there is every reason for more widespread and more precise understanding. Every one has personal contact with the physician at some time, and most people need the lawyer's advice periodically, but no similar direct need exists with regard to engineers. It is true that curiosity is aroused by contact of the public with some of the products of engineering and a mild interest excited, especially if the newspapers take up the matter. A more speedy ship on the transatlantic route, a new canal, an impressive waterpower project, a better automobile, a more economical electric lamp, a reduction in the cost of electricity, and a thousand other such evidences of engineering success, may attract attention, but this is not real public interest in engineering.

Real public interest in engineering will come, and with it active public cooperation, with benefits to both the public and the profession, only when there is public understanding that engineering is everybody's business, more or less, and most intimately the business of some who may not as yet have discovered the fact.

There are three broad divisions of proper public interest in engineering: first, the general one of acceptance and utilization of engineering methods, products and engineering services; second, that of investment of funds in enterprises, the soundness of which are primarily matters of engineering determination; and third, that of careers for young men as engineers in the profession.

Considering the first item, that of general use of the productions of engineering. It must be remembered that engineering is fundamentally creative, and that in ever-increasing degree new ideas and methods, new materials, new products, produced from both old and new materials, and better or more economical services are not only being produced regularly and systematically but that this will go on indefinitely in an ever-increasing degree. This is surely a matter of interest to everybody as possible consumers, but it is of especial interest to business men in every line of business, because trade conditions must necessarily change in consequence. Rather more important than this, as affecting the life, security, and prosperity of the

whole people, is the understanding of the engineering method, which is the broadening of scientific method to include economics, and then the adoption of this engineering method in all walks of life, in all sorts of affairs of business, and, especially, in matters of public relations and of government. This means, in effect, the elimination of untested mere opinion, of prejudice and mere tradition, of self-interest against general welfare and the substitution of facts and established principles, the substitution of that which is right and best, impersonally selected, impartially judged, and accepted only on proof of validity after survival over every known test.

The second rational basis for public interest in engineering is one of investment. This is especially important in countries where living conditions have advanced, as they have in the United States, to the point where it is possible for so large a part of the population to save some money from income, the surplus over living necessities being available for investment. It is, however, also important in other countries where correction of backward conditions may become possible by the adoption of similar methods, because the methods resulting in funds available for investment are the very methods that produce income for investments. Most of the important enterprises of the time, even those that superficially might seem to be mere exchange or trading affairs with none of the elements of creation of wealth in them, may in fact be founded on engineering, with values that have been increased by engineering. In many cases engineering has created a wholly new thing of real value, and so has actually produced wealth. No such undertaking is possible without capital, and the collection of capital through banks involves personal investments. The soundness of such activities as investments is a matter of engineering determination, subject, of course, to the usual reasonable safeguards of all business, classifiable as good management. It must therefore be recognized that engineering creates new investment opportunities that are natural consequences of the engineering method of creating wealth, not by robbing something to get something, but by true creation.

Finally, engineering as a career must be a matter of interest to the whole public concerned with the problem of most useful employment of its young men as a sociologic as well as an economic question, and especially is this the case with the parents of boys and the boys themselves. There is no more promising career conceivable today than that which engineering offers a young man of the right type. Everywhere, in every walk of life, are young men with the will to work, especially at jobs that are interesting and free of monotony, young men with the creative instinct, an inborn desire to make something. This is the same feeling that prompts the artist, the sculptor, the musician, the writer, the carpenter and the machinist, the tailor or the builder, the true business man and banker with pride in the business and its usefulness, rather than the profits that may be squeezed out of it, and fortunately this is the case with most that succeed. Every young man who feels that he would like to really do things, and there are many who would if only they were told the story of things to be done, is a possible recruit for the engineering army, and for him engineering is a possible career. There must, of course, be sufficient mental capacity to stand the test of training, to grasp the laws of nature that must be learned as part of the training, and to later on learn to play the game of solving the problems that are engineering as a sport where the game is the thing; a most absorbing, clean, satisfying and unselfish game, the game of making nature work for man as man works for himself and his fellow-man, the finest and greatest game in the world—engineering.—"Mechanical Engineering," Nov., 1926.

## BRUSHES FOR DIRECT-CURRENT MACHINES.

To many men responsible for the operation and maintenance of electrical machinery a brush is simply a piece of carbon, says Earl B. Stavely in the "Blast Furnace and Steel Plant." If a brush on a certain machine must be replaced, they assume that any spare brush will be satisfactory, or if no spare brush is available, they make one by cutting up any piece of carbon which may be at hand.

Satisfactory operation of direct current motors and generators, regardless of the class of service, can be obtained only by giving careful attention to the selection and maintenance of the brushes. Many commutation troubles which lead to severe sparking are due to the use of the wrong type or grade of brush. An improper application of brushes also increases the cost of machine maintenance, since it usually makes it necessary to replace brushes and re-surface commutators at frequent intervals.

For these reasons it is important that for each machine brushes should be selected which are suitable for use with that machine, and for the conditions under which it is operated.

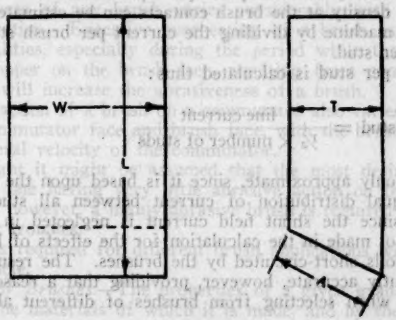



FIGURE 1.

Method of specifying brush dimensions.
A, arc of contact; L, length; T, thickness; W, width.

Brush manufacturers offer scores of different types and grades of brushes, each of which is designed to meet the requirements of certain definite conditions of service. If the properties of these different brushes are carefully considered, little difficulty should be encountered in obtaining a satisfactory brush for a given application. In some cases a satisfactory brush can be found only by trial of several grades. This is especially true when loading conditions are very severe or when difficult commutation problems are encountered. The grades which may well be tried under such conditions can be determined by analyzing the conditions of operation and by considering the properties of each type and grade of brush available.

An intelligent selection of brushes cannot be made unless one possesses some knowledge of the properties of brushes in general, and of the properties required to meet the demands of different applications. A portion of this article will be devoted, therefore, to a discussion of these topics.

*Properties of Brushes*—The properties of brushes which are of most importance in the majority of applications are: Current-carrying capacity; contact resistance; abrasive qualities. Other properties which must be given

consideration under certain conditions include: Hardness; density; coefficient of friction; lubricating qualities.

Brushes of different types and grades vary considerably in so far as these properties are concerned. If a brush is to be suitable for application in a given service it must possess the proper combination of these properties. Each of these will be discussed briefly in the following paragraphs.

*Current-carrying Capacity.*—The term "current-carrying capacity" means the current density at the contact between brush and commutator which should not be exceeded if serious overheating of the brush and commutator is to be avoided. Such overheating usually leads to severe burning and pitting of the brush face. This in turn causes sparking and the rapid wearing and burning of the brush.

Current-carrying capacity is expressed in terms of ampères per square inch of brush contact with the commutator. One grade of brush may have a current-carrying capacity of 60 ampères per square inch, while the capacity of a slightly different grade may be only 30. Serious overheating of the brushes will occur if this latter grade is applied, for example, in a machine with a contact density of 50 ampères per square inch.

The current density at the brush contacts can be estimated approximately for any given machine by dividing the current per brush stud by the brush contact area per stud.

The current per stud is calculated thus:

$$\text{Current per stud} = \frac{\text{line current}}{\frac{1}{2} \times \text{number of studs}}$$

This result is only approximate, since it is based upon the assumption that there is an equal distribution of current between all studs of the same polarity, and since the shunt field current is neglected in the calculation. Allowance is not made in the calculation for the effects of local currents in the armature coils short-circuited by the brushes. The result thus obtained will be sufficiently accurate, however, providing that a reasonable factor of safety is used when selecting from brushes of different allowable contact densities.

The contact area per stud is calculated thus: Contact area per stud = number of brushes per stud $\times$ A $\times$ W, where A and W are the dimensions in inches of the contact of each individual brush as indicated in Figure 1. Note that the distance across the brush face between tips is used in this calculation, not the thickness of the brush.

The current density is then—

$$\text{Current density at contact} = \frac{\text{current per stud}}{\text{contact area per stud}}$$

Commutation troubles and brush and commutator deterioration occur most frequently at the higher loads, hence in calculating contact densities for the purpose of brush selection the current at the maximum sustained load should be used.

*Contact Resistance.*—The term "contact resistance" means the electrical resistance of the contact between a brush and the commutator. This resistance varies with the current density at the contact, decreasing as the current density increases, and then increasing rapidly after the density becomes so high that the brush starts to glow, due to overheating. The contact re-

sistances of different types and grades of brushes vary over a rather wide range for the same contact density.

A low contact resistance is desirable, since it reduces the voltage drop and energy loss at the brushes, factors which affect the efficiency and temperature rise of the machine. A high contact resistance is desirable in order to improve commutation, since a high contact resistance reduces the local currents in the armature coils short-circuited by the brushes. As a compromise, therefore, a brush with a medium high contact resistance is often most satisfactory. In some cases a very high contact resistance is required in order to prevent commutation troubles, and a suitable brush must be selected, even though its use may have a marked effect upon the loss and temperature rise at the commutator.

*Abrasiveness.*—The abrasive qualities of a brush are determined by the amount of mechanical grinding action of the brush on the commutator. The abrasiveness of a brush depends upon the materials of which the brush is made, and not simply upon its hardness. Some grades of very hard brushes are only slightly abrasive, while some grades of very soft brushes are quite abrasive.

The abrasiveness of any brush varies considerably with different conditions of application. Excessive overheating of a brush usually increases its abrasive qualities, especially during the period while the brush is hot. A deposit of copper on the brush face, resulting from poor commutation or other cause, will increase the abrasiveness of a brush.

The grinding action of a brush on a commutator also varies with the condition of the commutator face and brush face, with the brush pressure, and with the peripheral velocity of the commutator.

At first thought it might be assumed that the most desirable grade of brush would be one having very little abrasive action on the commutator. In certain cases, however, a highly abrasive brush is required to avoid brush and commutator troubles.

The properties required for different brush applications will be discussed later in this article.

*Composition of Brushes.*—The properties of a given grade of brush are determined by the materials of which it is made, and by the methods used in its manufacture.

The materials used in brush manufacture include carbon, graphite, certain metals, and various kinds of abrasive material. These materials are properly mixed with a suitable binder and molded under high pressure into blocks. These blocks are then baked at a high temperature, after which they are finished to the desired shape and dimensions.

Pure carbon brushes are very hard, and have a low current-carrying capacity. They are not widely used at the present time.

A widely used type of brush is made of carbon, but the carbon is changed to artificial graphite during the process of manufacture. These brushes are called "graphitized-carbon" brushes. One method of manufacturing involves the heating of the material to a high temperature by electric current, and the brushes thus made are called "electro-graphitic brushes." They have a rather high current-carrying capacity when compared with brushes of untreated carbon.

Brushes made of pure graphite are quite soft, have a high current-carrying capacity, and have excellent lubricating qualities.

Certain grades of brushes are made of a mixture of carbon and graphite. The addition of the graphite increases the current-carrying capacity and improves the lubricating qualities. The properties of brushes of this type can be varied over a wide range by varying the relative amounts of carbon

and graphite used in their manufacture; hence brushes of this type are available in a variety of grades, each of which is suitable for definite conditions of application.

Metallized-graphite brushes are made of a mixture of graphite and certain metals, the amount of metal used varying with the purpose for which the brush is intended. Brushes of this type are used only in special applications where the required current-carrying capacity is so high that other types of brushes cannot be used. The purpose of the metal is to increase the current-carrying capacity to the necessary value.

The general properties of the different types of brushes are compared in Table I, the comparison being made with those of the electro-graphitic brush, since this type is most widely used.

TABLE I.

PROPERTIES OF DIFFERENT TYPES OF BRUSHES.

| Type. | Current-carrying Capacity. | Contact Resistance. | Hardness | Coefficient of Friction. |
|---|---|---|---|---|
| Carbon ..... | Low | High | Very hard | Medium |
| Graphitised carbon..... | Medium | Medium | Hard | Medium |
| Carbon-graphite ..... | Med. high | Low | Med. hard | Med. high |
| Graphite ..... | High | Low | Soft | High |
| Metal-graphite .. | Very high | Very Low | Hard | High |

The abrasive qualities of each type vary over a wide range depending upon the amount of abrasive material included in the materials of which the brush is made.

*Brush Properties Required.*—As noted previously, the conditions under which brushes are applied determine the type and grade of brush required. The brush properties required under certain conditions will be noted briefly here.

Brushes having some abrasive action on the commutator are required in case the mica is flush with the commutator. This is necessary in order that the brush can wear down the mica as rapidly as the copper bars are burned away by the current flow between brushes and commutator. High mica and serious sparking will result if non-abrasive brushes are used on machines in which the mica is not undercut. The degree of abrasiveness required depends chiefly upon the hardness of the mica and the natural tendency of the machine toward sparking.

Machines with slotted or undercut commutators should be equipped with brushes of very slight abrasive qualities. The only abrasive action required in this case is to keep the commutator face polished and clean.

Machines with slotted commutators should be provided with brushes of medium abrasiveness, however, in case severe overloads must be carried or in case commutation conditions are not good. This is necessary in order that the brushes will be able to wear away the burned surface of the commutator resulting from the heating and sparking which occur under the conditions mentioned.

A brush of low contact resistance should be used in machines which show no tendency toward sparking, even under overload. The loss and heating at the commutator can be reduced by the use of such brushes under these conditions.

In case sparking occurs, however, due to local circulating currents in the armature coils short-circuited by the brushes, it will be necessary to use brushes of rather high contact resistance if excessive sparking is to be avoided. Sparking due to this cause can often be greatly reduced by replacing the present brushes with a grade having a higher contact resistance.

The current-carrying capacity of the brush must also receive careful attention. The contact density, calculated as explained previously, should be checked against the rated current-carrying capacity of the brushes. Brushes of a higher carrying capacity should be used to replace existing brushes in case overheating of the brushes is found, and commutation conditions indicate the presence of considerable local current in the coils short-circuited by the brushes.

Other necessary properties will be noted in the following discussion of the selection of brushes for specific applications.

*Stationary Motors.*—Commutation conditions usually are quite good in these stationary motors. For these reasons a brush will usually prove satisfactory on motors with slotted commutators in this class of service if it possesses slight abrasive qualities. The coefficient of friction should be low if the commutator runs at a rather high peripheral velocity.

If the motors in this service have commutators with flush mica, the brushes should be more abrasive, and should have a higher contact resistance than those which are found satisfactory in motors with slotted commutators.

Both carbon-graphite and graphitized-carbon brushes are widely used in these applications of electric motors. Artificially lubricated brushes are frequently found desirable in case poor commutating conditions are encountered.

*Generators and Rotary Converters.*—Powerhouse generators and rotary converters are subjected to severe overloads which, while of short duration, occur at very frequent intervals. Rather severe sparking usually occurs during these periods of overload.

Brushes of rather high current-carrying capacity should be used on these machines if serious brush burning is to be avoided during overload. A brush of medium abrasive qualities will keep the commutator face in good condition, providing the commutator is slotted. A rather high abrasive brush will be required, however, if the mica is flush with the bars. Artificially lubricated brushes may well be used in this service in case commutation conditions are not good.

The required brush properties are summarized in Table II.

TABLE II.

| | |
|---|---|
| Stationary motors with flush mica | Medium current carrying capacity.<br>Fairly high contact resistance.<br>Fairly high abrasiveness. |
| Stationary motors with slotted commutators | Medium current carrying capacity.<br>Medium contact resistance.<br>Slight abrasiveness. |
| Generators and rotaries with flush mica | High current carrying capacity.<br>Fairly high contact resistance.<br>Fairly high abrasiveness.<br>Self-lubricating. |
| Generators and rotaries with slotted commutators | High current carrying capacity.<br>Medium contact resistance.<br>Slight abrasiveness.<br>Self-lubricating. |

*Brush Shunts.*—Brush shunts, or "pigtails," are provided on brushes to carry the current from the brush to the stud, or from the stud to the brush. These shunts are necessary in all cases where the brush slides in a box or holder. They are not necessary in case the brush is clamped to the holder in a fixed position, or in case special provision is otherwise made to conduct the current from brush to stud.

Shunts are necessary with sliding brushes in order to prevent burning of the holder and the brush sides. Figure 2 shows the desired current flow. If the shunt is defective the current will arc from the brush to the box, as indicated in Figure 2. This will cause the interior of the brush-box to be badly burned, in which case the brush may not move freely in the box. The burning of the brush sides, together with the burning of the box, will in time cause the brush to be too small for the box, and sparking troubles may result.

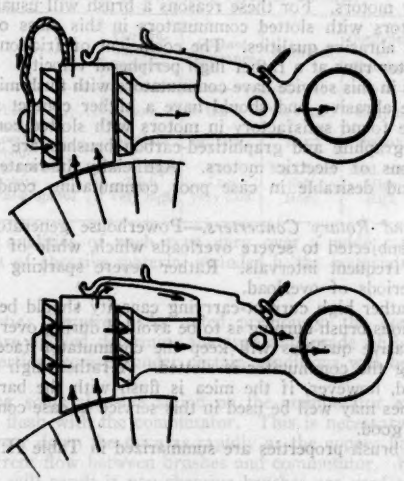


FIGURE 2.

Upper figure shows current flow from commutator to brush stud when brush is equipped with a shunt, or "pig tail," in good condition. Lower figure shows current flow when brush has no shunt, or shunt is in very poor condition. Current flowing through the spring causes it to lose its temper. Current flowing directly from brush to box of holder causes burning of sides of brush and inside of box.

A defective brush shunt will also cause a portion of the current to flow through the brush spring, as indicated in Figure 2. The resulting heating of the spring will cause it to vary its pressure on the brush. This may lead to serious unbalance in current distribution between brushes.

The shunt should be quite flexible in order that it may not interfere with

the movement of the brush in the box. The necessary flexibility is obtained by the use of shunts made of many strands of small wire.

The shunt is commonly attached to the brush by riveting it to the brush with a specially designed type of rivet, or by cementing it to the brush with a special current-conducting cement (see Figure 3). It is important that the shunt should be attached to the brush in such manner as to make a good mechanical and electrical connection, and that this connection be maintained in good condition.

The design of many machines is such that there is considerable possibility of accidental grounds or short circuits due to the brush shunts coming into contact with other parts of the machine. In such cases the shunts should be insulated, glass or lava beads proving quite satisfactory for this purpose. In some cases asbestos sleeving is found to be more satisfactory. This is especially true in the case of enclosed motors which are operated at high temperatures.

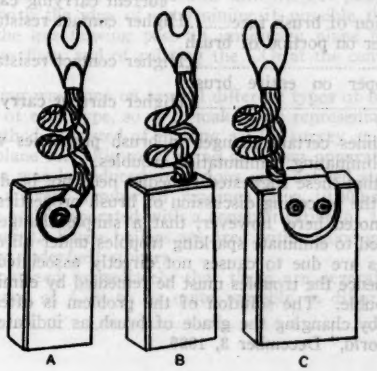



FIGURE 3.

Methods of attaching brush shunts to brushes. A, shunt held in contact with brush by a spun rivet; B, shunt attached to brush by copper cement which is tamped in hole in brush in which cable is inserted; C, shunt attached to copper saddle, this saddle being attached to brush by rivets.

The shunt should, of course, be so attached to the brush that no interference will be offered by it to the movement of the brush in the box or to the action of the brush spring.

*Commutation Troubles and Remedies.*—It is often found that a change in grade of brush will prove quite helpful in the solution of certain commutation and sparking troubles. Care must be exercised in making such changes, however, since a change in only one property may be required to eliminate existing troubles, while a change in other properties may lead to the development of other troubles.

TABLE III.

BRUSH CHANGES WHICH OFTEN PROVE HELPFUL IN THE SOLUTION OF COMMUTATOR PROBLEMS.

| Trouble. | Change to Brush Having— |
|---|---|
| Chatter, due to vibration | Greater lubricating qualities. |
| Chatter, due to rough commutator | Greater polishing action. |
| Chatter, due to dry commutator | Greater lubricating qualities. |
| High mica, due to excessive local currents | Higher contact resistance and greater abrasiveness. |
| High mica, due to sparking from other causes | Higher current carrying capacity and greater abrasiveness. |
| Burning and pitting of entire brush face | Lower contact resistance and higher current carrying capacity. |
| Burning of portion of brush face | Higher contact resistance. |
| Deposit of copper on portion of brush face | Higher contact resistance. |
| Deposit of copper on entire brush face | Higher current carrying capacity. |

Table III outlines certain changes in brush properties which may assist in reducing or eliminating commutation troubles.

It is believed that these suggested remedies need not be discussed in detail here in view of the preceding discussion of brush properties.

It should be noted here, however, that a simple change in brush grade cannot be expected to eliminate sparking troubles under all conditions. Most sparking troubles are due to causes not directly associated with the grade of brush used; hence the troubles must be remedied by eliminating the direct cause of the trouble. The solution of the problem is often made less difficult, however, by changing the grade of brush as indicated in the table.—"Mechanical World," December 3, 1926.

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## NOTES ON THE SAMPLING OF BOILER-FLUE GASES.

BY CHARLES GARRETT THATCHER,

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However expert the analysis of flue gas, the result cannot be better than the sampling. The percentage of $CO_2$ in a flue may easily range from 10 per cent near the walls to 15 per cent at the center. This being the case, careful attention to sampling methods is essential if $CO_2$ readings are to have much value. To find out the actual conditions for several standard types of boilers, the author has carried on an extensive practical investigation. The results are here presented in a form that will be helpful to engineers confronted with this problem in their own plants.

This article deals with the matter of obtaining a representative sample of flue gas, particularly with respect to its $CO_2$ content, from steam boiler passages.

A test engineer who inserts his gas-sampling tube into the center of the last pass of the boiler and obtains a true sample of the flue gases for a particular moment, is indeed fortunate. For it would seem that the time-

honored locality, the center of the last pass, though it might by accident yield a true average sample, generally will not, but will show a higher percentage of $CO_2$ than the true average. In nearly every instance the largest loss in boiler operation is that in the dry chimney gases.

It is evident that the result of the calculation of this loss can be correct only when a sample is obtained that is representative of the whole volume of gas at the particular moment. In a rough calculation of dry stack loss $CO_2$ alone may be used, leaving the other components out of consideration. The $CO_2$ content is generally considered a valuable index of the efficiency of combustion except for refined tests where $O_2$, O and CO are also determined.

Are your $CO_2$ meters and recorders telling you the average percentage of $CO_2$ in the gas as a whole? They are not if they draw a sample from the center of the last pass, as is most usual. But what difference does it make? Is not the whole volume of gas of nearly the same analysis throughout?

With these questions in mind the writer believed it worth while to explore with a sampling pipe and Orsat apparatus into the gas passages of several types of boilers with the idea of determining if possible the true average $CO_2$ content of the gas flowing past an imaginary plane perpendicular to the direction of gas flow, and of relating the $CO_2$ at the center of the plane to this average.

The experimenting was done on several different types of boilers and often on several boilers of each type, so as to make it as representative as possible. In general the method consisted of making a $CO_2$ survey at the last pass of the boiler, as on plane AA, Figure 1, using an Orsat apparatus for analysis. The sampling tube was introduced at various well-defined stations.

The results of a great number of readings on a 10,000-square-feet horizontal water-tube boiler, operated with a constant rating of 120 per cent, are illustrated in Figure 2. These boilers are fired with a good grade of Eastern bituminous coal by underfeed stokers.

It must be recognized immediately that there are several factors which cause the occurrence of $CO_2$ to be quite variable in different boilers and

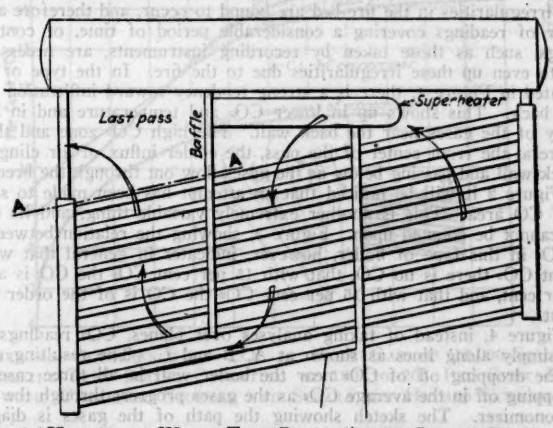


FIGURE 1.—HORIZONTAL WATER-TUBE BOILER (10,000 SQUARE FEET), SHOWING PLANE AA ON WHICH GAS SAMPLES WERE TAKEN.

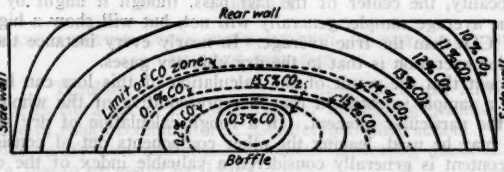


FIGURE 2.—OCCURRENCE OF $CO_2$ AND CO IN SAMPLES TAKEN ALONG PLANE AA (FIGURE 1). BOILER OPERATED AT 120 PER CENT RATING WITH UNDERFEED STOKER.

even in the same boiler at different times. Fire conditions, of course, have a great deal to do with this. The state of the grate and furnace walls, baffles, and side walls of the boiler also have a great effect, because of their control of the primary and secondary air supply. It is altogether conceivable, for example, that in a boiler whose side walls were in very bad condition, the percentage of $CO_2$ next to the wall might be 5 per cent instead of 10 per cent, as shown in Figure 2, or that in an extremely tight boiler the variation across the last pass might be much less than for the case shown.

The data here shown represent readings taken on several boilers in good average condition. However, it is certainly true that whether the variation in $CO_2$ across the pass is great or little it will follow the same general type of curve, and the problem of determining the representative gas sample is pretty much the same. An exceptional case might be cited, where one side of a boiler is in much worse condition than the other, causing the $CO_2$ curve to be unsymmetrical. In a tight boiler there is much less chance to make a serious error in obtaining a representative gas sample, aside from fire conditions.

Unevenness in the fire-bed makes it more difficult to obtain a true sample. These irregularities in the fire-bed are bound to occur, and therefore a great number of readings covering a considerable period of time, or continuous readings, such as those taken by recording instruments, are necessary in order to even up these irregularities due to the fire. In the type of boiler illustrated in Figure 1, there is a strong tendency toward infiltration of air at the back. This shows up in lower $CO_2$ and temperature and in higher velocity of the gases near the back wall. The high $CO_2$ zone and the CO area are at the front center of the pass, the colder influx of air clinging to the back wall and passing below as the gases flow out through the breeching.

In Figure 2 it will be noticed that an attempt has been made to show a limited CO area. This is another extremely variable thing, and its occurrence cannot be counted upon. Figure 3, showing the relation between CO and $CO_2$ in this type of boiler, however, indicates in general that with 14 per cent $CO_2$ there is no CO, that with 15 per cent $CO_2$ the CO is around 0.15 per cent, and that with 16 per cent $CO_2$ the CO is of the order of 0.5 per cent.

In Figure 4, instead of taking analyses over planes, $CO_2$ readings were taken simply along lines as shown at A, B and C. The resulting curves show the dropping off of $CO_2$ near the boiler wall in all three cases and the dropping off in the average $CO_2$ as the gases progress through the boiler and economizer. The sketch showing the path of the gases is diagrammatic only and does not show the fact that the economizer lies at right angles to the boiler. Thus the gases in this case have a rather long and winding

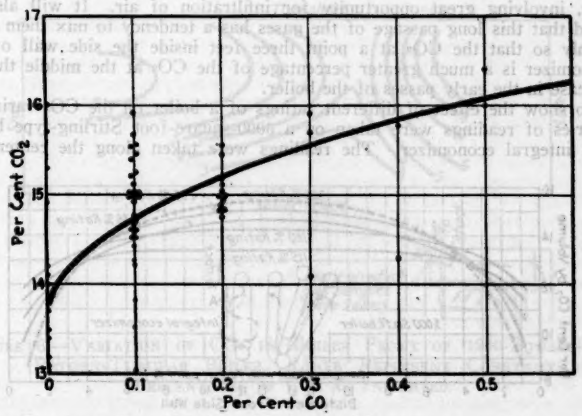


FIGURE 3.—RELATION BETWEEN THE OCCURRENCE OF $CO_2$ AND CO IN FIGURE 2.

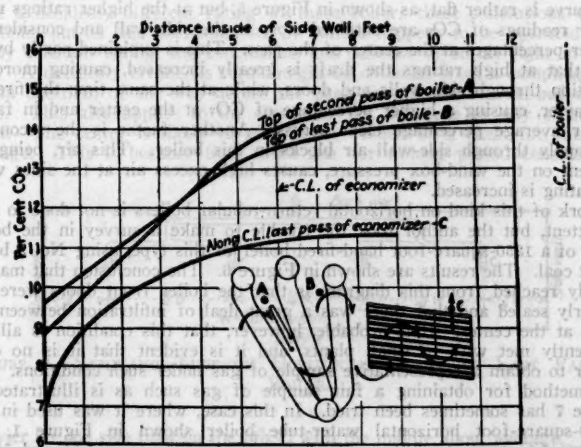


FIGURE 4.—OCCURRENCE OF $CO_2$ IN FLUE GAS OF 15,000-SQUARE-FOOT STIRLING-TYPE BOILER OPERATING AT 180 TO 200 PER CENT RATING WITH UNDERFEED STOKERS AND SEPARATELY HOUSED ECONOMIZER.

path, involving great opportunity for infiltration of air. It will also be noted that this long passage of the gases has a tendency to mix them thoroughly so that the $CO_2$ at a point three feet inside the side wall of the economizer is a much greater percentage of the $CO_2$ at the middle than is the case in the early passes of the boiler.

To show the effect of different ratings of a boiler on the $CO_2$ variation, a series of readings were taken on a 5000-square-foot Stirling-type boiler with integral economizer. The readings were taken along the center line

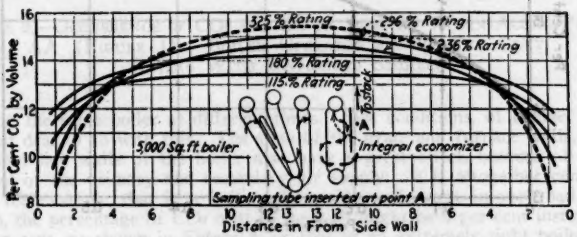


FIGURE 5.—VARIATION OF $CO_2$ WITH PER CENT OF RATING. SAMPLES TAKEN ALONG CENTER LINE OF LAST PASS. BOILER 5000 SQUARE FEET STIRLING TYPE WITH INTEGRAL ECONOMIZER AND UNDERFEED STOKER.

of the last pass of the economizer. It will be noted that for the low ratings the curve is rather flat, as shown in Figure 5, but at the higher ratings much lower readings of $CO_2$ are obtained close to the side wall and considerably higher percentages at the center of the pass. This is explained partly by the fact that at high ratings the draft is greatly increased, causing more infiltration through side walls and doors, while at the same time the fire-bed is heavier, causing a higher percentage of $CO_2$ at the center and in fact a higher average percentage clear across. Another factor is the secondary air supply through side-wall air blocks in this boiler. This air, being dependent on the wind-box pressure, causes high excess air at the sides when the rating is increased.

Work of this kind on horizontal return-tubular boilers is not done to such an extent, but the author took the trouble to make a survey in the boiler front of a 1250-square-foot hand-fired boiler of this type, using No. 2 buckwheat coal. The results are shown in Figure 6. The conclusion that may be quickly reached from this diagram is that the boiler front doors were not properly sealed and that there was a great deal of infiltration between the doors at the center. It is probable, however, that this condition is all too frequently met with in small plants, and it is evident that it is no easy matter to obtain a representative sample of gas under such conditions.

A method for obtaining a fair sample of gas such as is illustrated in Figure 7 has sometimes been tried. In this case, where it was used in the 10,000-square-foot horizontal water-tube boiler shown in Figure 1, the openings CC were along the center line and at points ⅛, ⅜, ⅝ and ⅞ of the distance across the last pass from the side of the wall.

### RESULTS CHECK CLOSELY WITH CALCULATED AVERAGES.

It was found from 48 readings, all taken at approximately 120 per cent boiler rating, that the results obtained checked almost exactly with the computed average $CO_2$. It was also found that in this set of boilers and at this rating

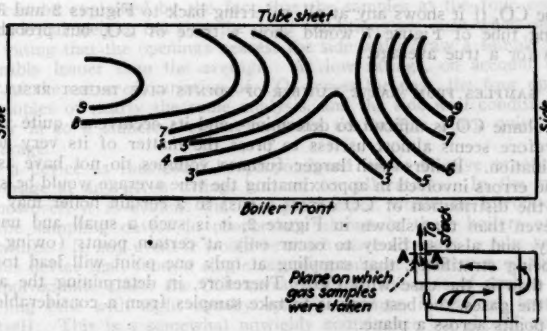


FIGURE 6.—VARIATION OF $CO_2$ IN BOILER FRONT OF 1250-SQUARE-FOOT RETURN-TUBULAR BOILER. LINES REPRESENT CONSTANT PERCENTAGES OF $CO_2$ AS INDICATED.

the calculated average $CO_2$, was about 90 per cent of the reading at the center of the last pass. It was further determined that a reading taken by introducing a sampling tube about one-fifth of the way across the center line of the last pass from the inside of the side wall would give the same result as the calculated average.

At a decidedly different rating of these boilers these conclusions would not hold, but as this particular set of boilers happens to be operated at a nearly constant rating, the conclusions are believed to be all right in their case. A decidedly erroneous result might be obtained from a sampling tube put in one-fifth of the way across the pass, on account of fuel-bed conditions. This also applies to the center of the last pass or any other one point. The center is likely to give better results as a basis for the true average. This is because fires often run for a long period considerably

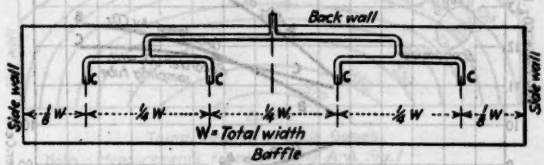


FIGURE 7.—SAMPLING TUBE ON HORIZONTAL PLANE IN LAST PASS OF BOILER.

heavier on one side than on the other, with the middle about average. However, it is the writer's opinion that sampling at only one point is bad practice, because the chances of that sample being representative of the whole are very small indeed.

The question naturally arises, How do these methods of sampling for $CO_2$ take care of the CO? Obviously, the method of going in from the side one-fifth of the way across the pass, will not give any indication of

the true CO, if it shows any at all, referring back to Figures 2 and 3. The sampling tube of Figure 7 would show a trace of CO, but probably not enough for a true average.

CO SAMPLES FROM LARGE NUMBER OF POINTS GIVE TRUEST RESULTS.

By volume CO is difficult to determine—and its occurrence quite variable. It therefore seems almost useless to press the matter of its very accurate determination. Boilers with larger furnace volumes do not have as much CO, and errors involved in approximating the true average would be smaller. While the distribution of CO across a pass in a certain boiler may be far more even than that shown in Figure 2, it is such a small and uncertain quantity, and also so likely to occur only at certain points (owing to the gases being stratified), that sampling at only one point will lead to worse errors than is the case with $CO_2$. Therefore, in determining the average CO in the gases, the best way is to take samples from a considerable number of points across a plane.

The type of sampling tube illustrated in Figure 7 was also introduced into a 15,000-square-foot Stirling-type boiler with integral economizer. The tube was put in the last pass of the economizer. This boiler is operated over a wide range of ratings. In Figure 8 curve A shows the average $CO_2$ taken at the center of the economizer last pass for various boiler ratings, B shows the calculated average from readings taken over the last pass, and C shows readings taken from the experimental sampling tube similar in proportion to that shown in Figure 8. It is evident that the sampling tube

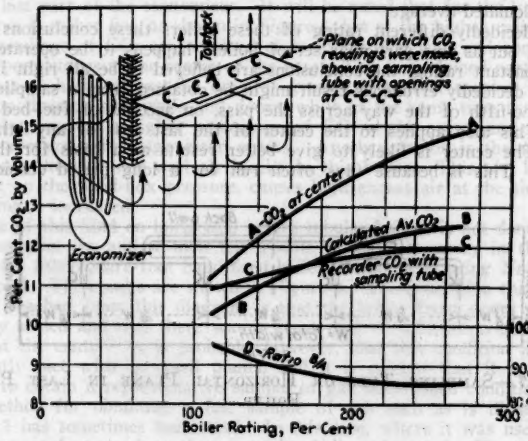


FIGURE 8.—$CO_2$ RELATIONS FOR LAST PASS OF 15,000-SQUARE-FOOT STIRLING-TYPE BOILER WITH INTEGRAL ECONOMIZERS.

gives fairly accurate results within the range of 140 to 240 per cent of rating, which is the usual range for these boilers, but that the sampling tube gives a reading a little too high at the lower ratings and a little too low at the higher ratings.

This can be explained by the fact that the samples at the four openings are given equal weight, but that the draft conditions are such when operated at high rating that the openings nearest the side walls draw a sample that is considerably leaner than the average. At low ratings, on account of the flatness of the curve of variation of $CO_2$ across the pass, the four openings draw samples of nearly the same analysis, and the side-wall condition does not enter in to a sufficient extent to bring the sample down quite to the true average.

Such a device as this could be proportioned so as to give practically a true average at any given rating, but not for all ratings. In the instance cited, however, it is evident that no great error would be introduced by using the sampling device in proportions shown in these boilers at their usual ratings. In any event it would be much more accurate to take readings from this device than from a single sampling tube at the center of the pass.

To approximate closer the $CO_2$ average across the pass of such a boiler, a sampling tube with eight equally spaced openings, instead of four, suggests itself. This is a somewhat unwieldy contrivance but should give satisfactory results. The use of a sampling tube with four equally spaced openings, as illustrated in Figure 7, but with the outside openings slightly smaller than the inside, might prove somewhat more accurate, since it would give a little less weight to the samples taken near the wall. When using tubes with more than one opening, provision for blowing out the soot should be made. To make the blowing positive, the pipes to each sampling point might better be separate, each with its own connection for blowing, and joining into a common header outside the boiler wall.

In computing an average $CO_2$ content for a certain plane, all gas-sampling stations on that plane should not be given the same weight since the velocity of the gas varies and also the temperature. In order to find out the variation in temperature, a considerable number of readings were made with a thermocouple and pyrometer in the last pass of a horizontal water-tube boiler, as shown in Figure 1. To obtain an indication of the variation in velocity, a pitot tube was inserted and readings made on a differential draft gauge. The pitot tube had two branches, one projecting upstream and one

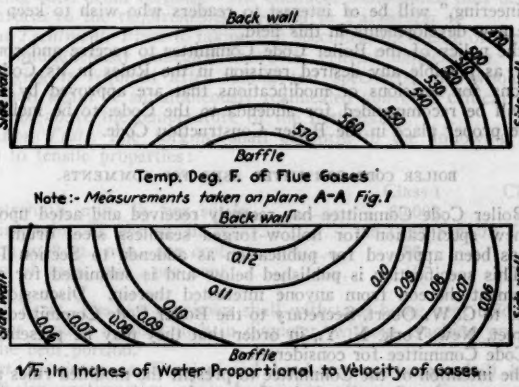


FIGURE 9.—VARIATIONS IN $CO_2$ AND GAS VELOCITY ALONG PLANE A, FIGURE 1.

downstream, and therefore the reading on the draft gauge was twice the velocity head expressed in inches of water. Determining the square root of $h$ gave a value proportional to the velocity at the various points in the plane. Figure 9 shows the results of these experiments in determining temperature and the square root of $h$.

A weight factor was worked out for each station in the plane, this factor being the square root of $h$ divided by the square root of T, where T is the absolute Fahrenheit temperature. The weighted average $CO_2$ is 13.105, while the simple numerical average of CO readings for the plane is 12.95. Nothing about the whole proceeding warrants making this attempt at an extremely accurate $CO_2$ average, although it would generally be found that the weighted average, since it gives more weight to the stations near the middle of the pass where the velocity and $CO_2$ are both higher, would be higher than the numerical average. In case the setting were in bad condition and the $CO_2$ near the walls fell away off, this difference would be greater.

In working out weighted $CO_2$ averages for the two other types of boilers shown in Figures 4 and 5, the writer left temperature out of consideration and used only the square root of $h$ as a modifying factor. The same conclusion was reached as before, namely, that the weighted average was very close to the numerical average and that the effect of varying velocities and densities across a pass in a boiler should not be considered.—"Power," November 23, 1926.

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## HIGH-PRESSURE FORGED DRUMS TO BE COVERED BY A. S. M. E. CODE.

The existence of the extremely high-pressure boilers with forged drums and the probability that other such boilers will be constructed, has been recognized by the A.S.M.E. Boiler Code Committee in a proposed addition to the Boiler Construction Code. The following, which is quoted from a report of the Code Committee published in the November issue of "Mechanical Engineering," will be of interest to readers who wish to keep in touch with the new developments in this field.

It is the policy of the Boiler Code Committee to receive and consider as promptly as possible any desired revision in the Rules in its Codes. Any suggestions for revisions or modifications that are approved by the Committee will be recommended for addenda to the Code, to be included later on in the proper place in the Boiler Construction Code.

### BOILER CODE COMMITTEE ASKS FOR COMMENTS.

The Boiler Code Committee has recently received and acted upon a suggested new specification for hollow-forged seamless steel drum forgings which has been approved for publication as addenda to Section II of the Code. This specification is published below and is submitted for criticisms and comment thereon from anyone interested therein. Discussions should be mailed to C. W. Obert, Secretary to the Boiler Code Committee, 29 West 39th Street, New York, N. Y., in order that they may be presented to the Boiler Code Committee for consideration.

It is the intention of the Committee to present the modified rules as finally agreed upon to the Council of the Society for approval as an addition to the Boiler Construction Code. Upon approval by the Council, the revisions

will be published in the form of addenda data sheets, distinctly colored pink, and offered for general distribution to those interested, and included in the mailings to subscribers to the Boiler Code interpretation data sheets.

## PROPOSED SPECIFICATIONS FOR HOLLOW-FORGED SEAMLESS STEEL DRUM FORGINGS.

1. *Process.* The steel shall be made by either or both the following processes: open-hearth or electric furnace.

2. *Discard.* Sufficient discard shall be made from the top and bottom of each ingot to secure soundness in the portion used for the drum forging.

3. *Forging.* The forging shall be made from a solid cast ingot, punched, bored, or hot trephined.

The resultant wall of ingot shall be reduced in thickness at least one-half by forging on mandrels.

4. *Chemical Composition.* The steel shall conform to the following requirements as to chemical composition:

| | | Class 1 | Class 2 |
|---|---|---|---|
| Carbon, maximum, per cent | | 0.35 | 0.45 |
| Manganese, per cent | | | 0.30–0.65 |
| Phosphorus, maximum, per cent | Acid | | 0.04 |
| | Basic | | 0.035 |
| Sulphur, maximum, per cent | | | 0.04 |

5. *Ladle Analyses.* An analysis of each melt of steel shall be made by the manufacturer to determine the percentage of the elements present. This analysis shall be made from a test ingot taken during the pouring of the melt. The chemical composition thus determined shall be reported to the purchaser or his representative and shall conform to the requirements specified in Paragraph 4.

6. *Check Analyses.* As a substitute for check analyses in the general rules, the purchaser may make an analysis from a broken tension-test specimen representing each drum forging. The chemical composition thus determined shall conform to the requirements specified in Paragraph 4.

7. *Heat Treatment.* Prior to taking test specimen, the whole of the forging shall be simultaneously annealed above its critical temperature. If additional forging is required after taking test specimens, the whole of the forging shall again be simultaneously reannealed above its critical temperature, but not above the temperature of the first anneal.

8. *Tension Tests.* The forging shall conform to the following requirements as to tensile properties:

| | Class 1 | Class 2 |
|---|---|---|
| Tensile strength, min., lb. per sq. in. | 60,000 | 75,000 |
| Yield point, min., lb. per sq. in. | 0.5 Tens. Str. | 0.5 Tens. Str. |
| Elongation in 2 in., min., per cent | 26 | 24 |
| Reduction of area, min., per cent | 42 | 38 |

9. *Bend Tests.* The test specimen shall withstand being bent cold through 180 degrees around a pin 1 inch in diameter, without cracking on the outside of the bent portion.

10. *Test Specimens. a.* Tension and bend-test specimens shall be taken from full-size prolongation of each forging after annealing, as provided for in Paragraph 7.

(*b*) One tension-test specimen shall be taken from each end of the forging. The axis of the specimens shall be located midway between the inner and outer surfaces of the wall parallel to the axis of the forging, the two specimens being taken from diagonal corners of an axial plane. Tension-test specimens shall conform to the dimensions shown* in Figure 1. The ends shall be of a form to fit the holders of the testing machine in such a way that the load shall be axial.

(*c*) One bend-test specimen shall be taken from the end of the forging corresponding to the top of the ingot. The axis of the specimen shall be in a diametral plane perpendicular to the axis of the forging. The bend-test specimen shall be 1 inch by ½ inch in section, with edges rounded to 1/16 inch radius.

11. *Number of Tests.* (*a*) Two tension tests and one bend test shall be made for each forging.

(*b*) If any test shows defective machining or develops flaws it may be discarded and another specimen substituted.

(*c*) If the percentage of elongation of any test specimen is less than that specified and any part of the fracture is more than ¾ inch from the center of the gauge length, as indicated by scribe scratches marked on the specimen before testing, the specimen shall be discarded and another taken.

12. *Retests.* If the results of the physical tests of any forging do not conform to the requirements specified, the manufacturer may reanneal the forging and retests shall be made as specified.

13. *Permissible Variation.* The thickness of the parallel wall of each forging shall not be less than that specified. If the thickness of a portion of the wall of the forging is less than that specified, due to accidental or unavoidable irregularity of bore, the forgings may be accepted by the purchaser, provided such irregularity will not require lowering of the allowable working pressure below that for which the drum is designed.

The overweight limits are considered a matter of contract between the manufacturer and the purchaser.

14. *Finish.* The forging shall be free from injurious defects and shall have a workmanlike finish.

15. *Marking.* Each forging shall be legibly stamped by the manufacturer at each end with the name of the manufacturer, manufacturer's test identification number, and minimum tensile strength specified for the class of forging. The manufacturer's test identification number shall be legibly stamped on each test specimen.—"Power," December 7, 1926.

## CHARACTERISTICS OF CENTRIFUGAL PUMPS.

By F. Johnstone-Taylor.

In the case of centrifugal pumps there are certain fundamental variables, such as head, discharge, efficiency, and brake horsepower, which when plotted on section-paper constitute the characteristics of the pump. These curves may in the case of cheap stock-pattern pumps be somewhat misleading, but in the case of well-designed and constructed units they provide, when analyzed, a good deal of information about the performance of any individual pump which otherwise would not be readily apparent. The relations between the aforementioned values are fixed absolutely for a given

* Fig. 1, not reproduced here, shows a cylindrical test piece with the central portion turned down to a uniform diameter of 0.505 inch. This portion ends in fillets of radius not less than ½ inch, the fillet centers being 2¼ inches apart. A 2-inch gauge length is laid out for measuring elongation.

impeller, although the designer, by so proportioning the diameter of the impeller, its vane angles, and their curvature, etc., can give to a pump widely different characteristics as might be deemed necessary by the operating conditions of that pump. As a rule, characteristic curves are based upon a constant speed, and they show the relationship between the head, discharge, efficiency, and brake horsepower at this speed, although if any one of the fundamental values be kept constant and the others varied, other curves may be obtained which are of use under certain conditions. Typical curves are given in Figure 1 of a single-stage pump designed for a normal

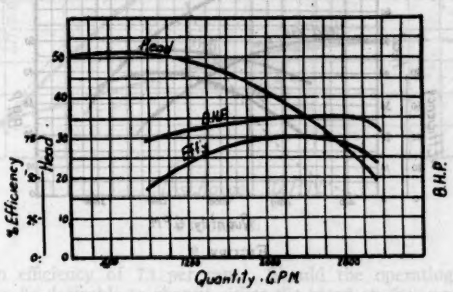


FIGURE 1.

duty of 2400 gallons per minute against a head of 35 feet at a speed of 835 R.P.M. Such a curve would be plotted from data obtained on the pump test-bed by running up the pump to speed with the discharge valve closed, and then obtaining a series of readings of the head, discharge, and power absorbed for successive openings of the valve. Thus it will be noted that as the head is decreased the efficiency rises and the horsepower absorbed increases. They both reach their maximum value when the pump is discharging at its rated capacity, after which the efficiency rapidly falls off. The horsepower curve will be seen to rise a small amount, a condition of affairs which is met by providing a motor of about 10 per cent to 15 per cent in excess of the power absorbed by the pump under normal conditions. Thereafter the brake horsepower curve falls, indicating that the pump is of the non-overloading variety. It should be noted that if such curves are to have real value as to their indication of what any particular pump is capable of, they must be plotted from data obtained under test for any given pump, although curves estimated from theoretical data do often correspond quite closely with what the pump will do in practice. Curves obtained from test-bed data, as indicated, represented the performance of any pump until such time as the proportions of the impeller become changed by wear.

*Effects of Speed Increase.*—It is usual, though not essential, to run centrifugal pumps at constant speed. In any case they are well suited to operation by alternating current motors, which are essentially constant-speed machines. The head in the case of a centrifugal pump varies as the square of the speed and the discharge directly as the speed, and quite irrespective of either the type or the proportions of the impeller. This condition may cease to be true if the velocity at entry to the impeller becomes high enough to induce chokage, but as a rule such velocities considerably exceed those provided for by the designer. Referring to Figure 2, this shows curves for the same

pump as is represented by Figure 1, but running at 1000 R.P.M. The curves in Figure 2 can also be constructed from the data from which those in Figure 1 are plotted, bearing in mind that the discharge increases directly as the speed and the head as the square of the speed, and both calculations are quite simple. For instance, take the point where the pump is discharging 1600 gallons per minute at 835 R.P.M.; at 1000 R. P. M. the discharge will

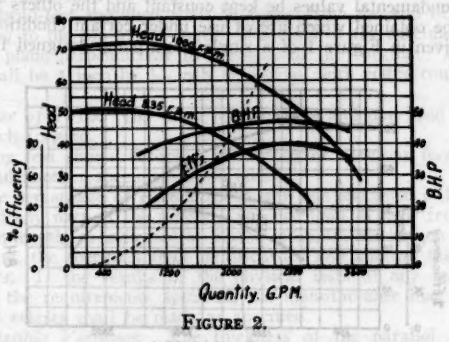


FIGURE 2.

be $1600 \times 1000/835 = 1915$ gallons per minute, and the head indicated in Figure 1 as 46 feet becomes $46 \times 1000^2/835^2 = 58$ feet. The efficiency of the pump, again, when discharging 1600 gallons per minute is 68 per cent, the maximum efficiency being 76 per cent, therefore the relative efficiency is $68/76 = 90$ per cent (nearly). At 1000 R. P.M. the maximum efficiency is 80 per cent, so that at the particular point in question the efficiency can be estimated at $0.80 \times 0.90$, or, say, 72 per cent.

As regards the efficiency curve, after each change of speed the relative efficiency of each point of the head-discharge curve remains the same as previously, and by relative efficiency is meant that at any particular point in percentage of the maximum at the same speed. After a change of speed the actual efficiency may be slightly above or below that which previously obtained, according to the design of the impeller.

As regards horsepower absorbed, as the discharge varies with the speed and the head as the square of the speed, the horsepower varies with the cube of the speed, so that with this information, should the full discharge not be obtainable from a large pump on the test bed, it is possible from plotted head, capacity, and efficiency curves, to calculate this with a fair degree of accuracy for other speeds or powers. Or, to sum up, if a pump characteristic, or even one set of conditions, is known, the effect of changes in speed or of impeller diameter can be estimated from the following relation, which holds good at constant efficiency:—

$$\frac{N_1}{N_2} = \frac{Q_1}{Q_2} = \frac{\sqrt{H_1}}{\sqrt{H_2}} = \frac{\sqrt[3]{H.P._1}}{\sqrt[3]{H.P._2}},$$

where $N_1$ and $N_2$ are respective speeds, $Q_1$ and $Q_2$ respective discharges in gallons per minute, and $H_1$ and $H_2$ head in feet for the respective conditions.

In Figure 3 the solid lines show actual test curves of a pump at 1750 R.P.M. The operating conditions are 1200 gallons per minute at 157-foot

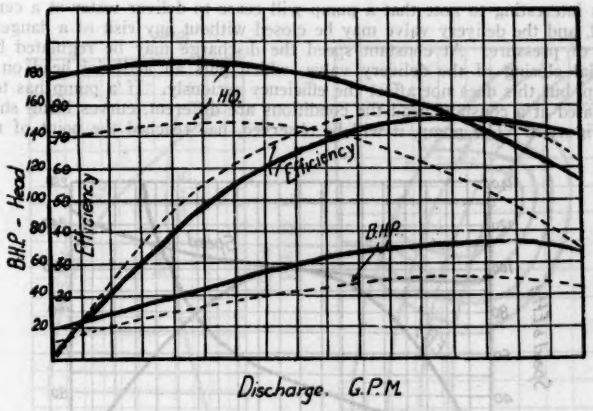


FIGURE 3.

head with an efficiency of 73 per cent. Should the operating conditions change, it may be desirable to change either the speed or the impeller of the pump. Assume, for instance, the head reduced to 118 feet, but that the same discharge is required. By reducing the speed to 1560 R.P.M. the H Q curve would be changed to that indicated by the dotted lines, the other values being similarly indicated.

*Change of Speed, Constant* K.—In the case of a pump driven by, say, a steam turbine or a variable-speed motor, the speed may be estimated for a given delivery at a given head. The pump previously referred to has a normal duty of 2400 gallons per minute at 35-foot head. It is required to deliver this quantity at 60-foot head, so that an increase of speed will be necessary. A constant K has now to be determined, because H varies with $Q^2$ and $H = Q^2 K$. This is the equation of a parabola passing through the intersection of the H and Q axes, and in this particular instance it must pass through point H = 60 and Q = 2400. Therefore—

$$60 = K\ (2400)^2 = K = \frac{60}{5{,}760{,}000} = 0.00001059$$

$$\therefore\ H = 0.00001059\ Q^2.$$

Then for successive values of Q, H must be calculated giving the dotted line in Figure 2 where it crosses the head curve at normal capacity at 2000 gallons per minute. As Q varies directly as S the speed, the required speed for a delivery of 2400 gallons per minute at 60-foot head is equal to $835 \times 2400/2000 = 1000$. The plotting of the head-capacity or "constant-efficiency" curve is rather a laborious process, but it can be greatly simplified by the use of logarithmic paper, because a parabola passing through the origin thereof becomes a straight line.

*Constant Speed and Constant Head.*—It has already been pointed out that decreasing the head calls for an increase of power if the speed be kept constant, and a reasonable overload capacity should be provided for in the motor.

It is interesting to note that a pump will cease to deliver water at a certain head, and the delivery valve may be closed without any risk of a dangerous rise of pressure. At constant speed the discharge may be regulated by a partial closing of the delivery valve, which puts an artificial head on the pump, but this does not affect the efficiency seriously. If a pump has to be operated at a constant head the conditions are different, curves being shown in Figure 4. The pump, it will be observed, has reached the point of non-

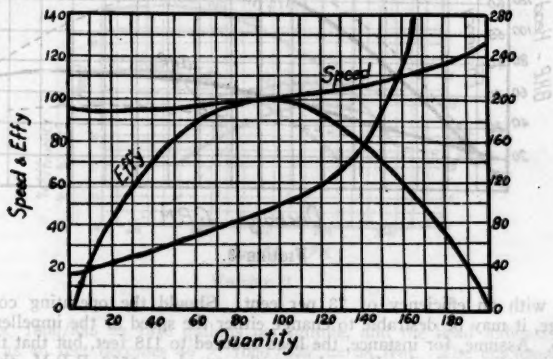


FIGURE 4.

NOTE.—In this diagram the variation of speed, discharge power, and efficiency are expressed as a percentage of the normal conditions.

delivery when the speed has dropped to 95 per cent of the normal, so that it is not generally desirable to regulate the delivery of the pump below its normal by speed regulation if the head is constant, for it is evident that it will be in a condition which will be more or less unstable, since with a small decrease in speed it fails to deliver water. Under most average pumping conditions the head would remain constant within quite small limits, so that a constant-speed pump suits most installations. There are, however, such instances as dock pumps, sinking pumps, and fire pumps which call for a wide variation of head, to meet which a self-regulating pump may be installed. The orthodox type of centrifugal pump is not self-regulating, although to design a pump to give a self-regulating characteristic curve is not a matter of any difficulty. Yet any such alteration from the normal towards self-regulation means a decrease of efficiency at the point of normal duty. This cannot be avoided owing to the nature of the method by which the alteration is obtained. This property, therefore, with the orthodox type of pump has its limitations. The object of a self-regulating pump is to avoid the provision of a large margin of power over and above that required for normal duty, so as to keep down the first cost and upkeep charges of the installation.

Typical curves of such a pump are given in Figure 5. Some types of pump are specially designed to be automatically self-regulating. The Rees Roturbo pump, for instance, with its specially designed pressure chamber, is always discharging into a constant pressure, and the turbine which is in

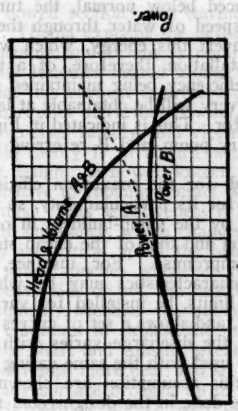


FIGURE 6.

CENTRIFUGAL PUMPS.

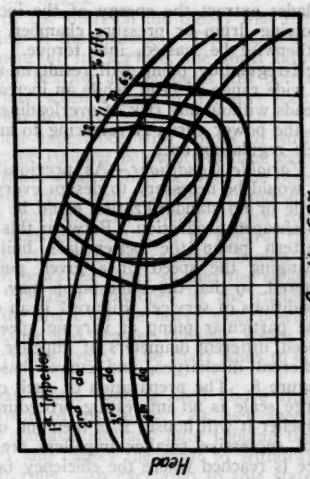


FIGURE 8.

CENTRIFUGAL PUMPS.

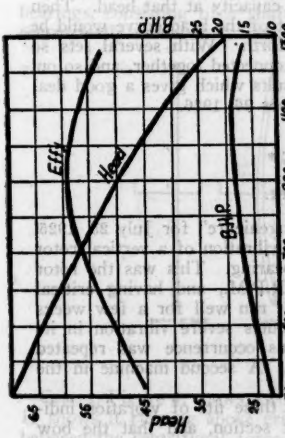


FIGURE 5.

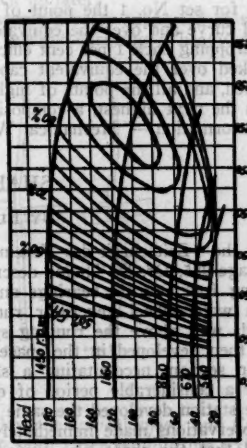


FIGURE 7.

series with the impeller is always discharging from constant pressure. The result is that if the head varies or is reduced below normal, the turbine blades extract the energy of the increased speed of water through the revolving drum or pressure chamber, and convert this energy, which would otherwise be wasted, into torque. The installation, therefore, of a truly self-regulating pump will result in a high efficiency being maintained over a wide range of head, while an increased delivery will be obtainable at lower heads without any risk of overloading the motor. This is indicated in Figure 6, the power curve A referring to an ordinary pump, and B referring to a self-regulating one.

*Variable Conditions.*—As previously stated, for the maximum efficiency it would be necessary to design every pump for its particular duty, so that little or no standardization could be effected by the pump-builders in order to cheapen production. Between this condition and that of the cheap stock-pattern pump the better-class builders compromise. For instance, by changing the speed of a given pump its characteristics may be wholly altered, so that the same pump may, within limits, be installed for various conditions of service. Figure 7 is an example, and shows a set of curves for one particular pump at varying speeds. As the discharge varies with the speed, different diameters of impeller may be used with the same casing for different discharges, and curves based on this assumption are shown in Figure 8. The preparation of such curves as done in the design-room to a large scale is an interesting procedure. A casing is fitted with the largest impeller it will house; it is then run under test, and its curves plotted. Several successive smaller impellers are dealt with in the same way until a size is reached where the efficiency falls off very rapidly. The head curves would all be then plotted on one sheet, and by reference to the efficiency curves for each impeller points are marked off on the final sheet to correspond to the different efficiencies for the various impellers. That is to say, for set No. 1 the point of highest efficiency would be marked on the head curve and over the equivalent position of capacity at that head. Then by dropping back 1 per cent efficiency that point of the head curve would be marked over the equivalent capacity, and so forth. With several sets so plotted, and all the points of highest efficiency connected together, and so on, with the next highest, a composite diagram results which gives a good deal of information.—"Mechanical World," November 26, 1926.

## SHAFT RUBBING.*

By Burt L. Newkirk.

In the "Zeitschrift des Vereines deutscher Ingenieure" for July 25, 1925, on page 985, Dr. D. Thoma described a case of vibration of a vertical rotor having four journal bearings and one thrust bearing. This was the rotor of a water-wheel generator running at 600 R.P.M., and having critical speeds well above the running speed. The unit ran well for a few weeks and then developed in the space of a few minutes severe vibration in its lower section, necessitating a shut-down. This occurrence was repeated after a considerable period of quiet operation. A second machine in the same station developed the same trouble.

Observations made immediately after one of these fits of vibration indicated that the shaft was bowed in its lowest section, and that the bow disappeared within a few minutes after shut-down. It had been observed

* "Proceedings of the American Society of Mechanical Engineers."

previously that the bearing at the upper extremity of the lowest shaft section (the bowed section) ran warmer than the others.

Dr. Thoma explains the deformation of the shaft on the theory that it is in unstable equilibrium as regards heating in the bearings. That is, if the journal becomes heated along one element of its surface, the shaft bows so that this element is pressed against the bearing and becomes still hotter, which results in further bowing, etc. Thus, when a cycle of this sort starts it progresses rapidly, and violent vibration develops.

This theory is in accord with common experience with rotors of stiff-shaft design. It is well known that a rotor running below its critical speed is sensitive to rubbing contact with the shaft at any point between the outer bearings. This sensitiveness is due to the cycle just described, complicated by the unbalance, which changes as the shaft distorts. When a rotor runs below its critical speed the heavy side runs out, and it is in general this side which touches whenever rubbing starts. Whether rubbing begins on the heavy side or not, the heat developed bows the shaft so that the rubbed side quickly becomes the heavy side. The heat developed by the rubbing then continues to expand the metal of the shaft on the heavy side, bowing out the shaft and causing greater unbalance and heavier rubbing, which in turn develops more heat, and so on in a vicious cycle. The shaft tends to dig into any surface brought up to it.

It is perhaps not so well known that shafts running well above their critical speeds are not subject to this form of instability. At such speeds the shaft bows so that the heavy side is concave, and any light rubbing that occurs heats the light side (the outside) and causes the shaft to withdraw itself from the surface against which it rubs. An exception to this statement must be made of the case of shafts running true and in perfect balance, and of shafts having a ridge or prominence which rubs first, irrespective of the state of balance. These cases will be taken up later.

To visualize the effects described above, assume a rotor supported at two bearings, consisting of a shaft of negligible weight carrying a heavy disc eccentrically mounted.

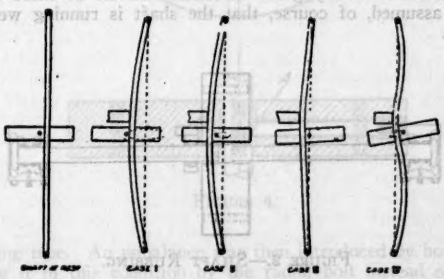

FIGURE 1.

Case I, Figure 1, shows a rotor running below the critical speed, and Case II shows it running above critical speed. In both cases equilibrium exists between the centrifugal force due to the whirling mass and the elastic restoring force of the shaft. In both cases the rubbing heats and expands the elements on the outside of the shaft, weakening its elastic restoring force. In Case I equilibrium can be re-established by increasing the radius of

whirl; but in Case II it can be re-established by reducing the radius of whirl. This can be verified by setting up the algebraic condition of equilibrium for each case. It should be directly evident, however, that in Case I, where the center of mass is outside the shaft, the ratio of elastic restoring force to centrifugal force increases with increasing radius of whirl, and in Case II this ratio increases with decreasing radius of whirl.

In Case I, if the solid body is brought up to the whirling shaft so as to produce light rubbing, the shaft should bend more and try to dig into the surface, getting progressively rougher. In Case II, if the solid body is brought up so as to be rubbed by the whirling shaft, the radius of whirl should decrease, and the shaft should withdraw itself from the solid body. The rotor would then run in better balance than before.

If now in Case II the solid body be brought nearer still, the shaft should again withdrew itself, and this cycle should continue until the shaft runs in balance with its center of gravity on the line of bearing centers (Case III). The shaft would, however, still be whirling owing to the eccentric mounting of the disc. The bend in the shaft shown in the figure is due to the heating up of one side of the shaft, which is under no bending stress.

If now the solid body still moves towards the shaft, the rotor would continue to withdraw until finally the shaft ran true at the rubbing surface, but out of balance by reason of the eccentric mounting of the disc (Case IV).

Since the shaft has now come to run true at the rubbing surface it can no longer withdraw from the surface, but if rubbing continues the shaft would not tend to dig in with one element getting more and more heated. Any further heating of this one element would increase the distortion of the shaft shown in Case IV, and so cause the element on the opposite side of the shaft to rub. Thus the rubbing would pass rapidly from one element of the shaft to another, and there would be no further progressive bending of the shaft.

The shaft running true and in perfect balance would also behave in this way. It could not withdraw from a rubbing surface, but different elements of the shaft would receive the rubbing successively, so that no considerable bowing would occur in any given direction. In all of these cases except Case I it is assumed, of course, that the shaft is running well above its critical speed.

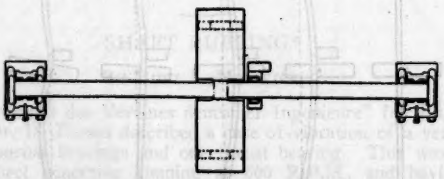

FIGURE 2.—SHAFT RUBBING.

This phenomenon was studied experimentally in the course of a study of shaft behavior made in the General Electric Company's research laboratory some time ago. A rotor shown in Figure 2 was first balanced carefully, and then an additional balance-weight was inserted to throw the rotor out of balance by a known amount at a known point. The rotor was then run at various speeds, and a surface was set so that the shaft rubbed slightly against it. When the rotor ran below its critical speed the result was in all

cases that rubbing increased in intensity, building up more rapidly the more severe the rubbing. When the shaft was examined after the run the mark due to rubbing was found on the heavy side (Figure 3). When the speed was above the critical speed of the rotor, the rubbing, if not too severe at the start, did not build up, but cleared itself without removal of the rubbing surface. If the rubbing was too severe with the shaft running above its

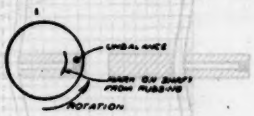


FIGURE 3.

critical speed, the frictional resistance to rotation would reduce the speed below the critical, and vibration would then build up.

To supplement this test the unbalance was produced in a different way, which, however, should make no difference in the behavior of the rotor. Figure 4 shows a disc mounted so that pressure can be exerted on the shaft at A, causing the shaft to bow. The rotor was first nicely balanced with the

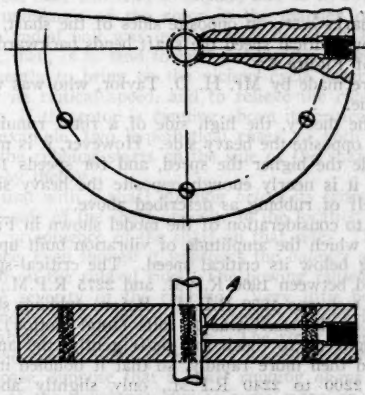

FIGURE 4.

shaft running true. An unbalance was then introduced by bowing the shaft and holding it in this condition by the radial bolt instead of inserting an additional balance-weight. The unbalance was introduced in this way because it seemed hard to believe that a bowed shaft carrying balanced discs would bend in the reverse direction when running above its critical speed (Figure 5).

This rotor was not used for rubbing tests because its shaft was of small diameter, and it seemed likely that the heat due to rubbing would distribute itself rapidly and have little bowing effect. However, the shaft was marked to determine the high side when running at speeds below and above the

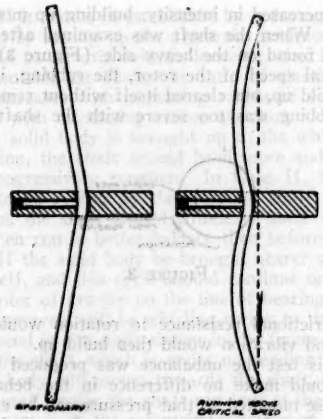


FIGURE 5.—SHAFT RUBBING.

critical. These marks were on opposite sides of the shaft, and showed that when running above critical speed the shaft bends backward as shown in the second diagram of Figure 5.†

These tests were made by Mr. H. D. Taylor, who was working with the author at that time.

According to the theory, the high side of a rotor running above critical speed is not quite opposite the heavy side. However, it is more nearly opposite the heavy side the higher the speed, and for speeds 10 per cent above the critical speed it is nearly enough opposite the heavy side to permit the shaft to clear itself of rubbing as described above.

Returning now to consideration of the model shown in Figure 2, Figure 6 shows the rate at which the amplitude of vibration built up in one test with the model running below its critical speed. The critical-speed vibration of this rotor occurred between 1960 R.P.M. and 2275 R.P.M., and the running speed during this test was 1800 R.P.M. Before rubbing started the amplitude of vibration of the shaft was 10 mils (total motion). When a brass surface was brought up to touch the shaft gently, the amplitude built up at first slowly and then more rapidly, so that it doubled in 4 minutes. At speeds of from 2200 to 2240 R.P.M., only slightly above the critical, repeated rubbing failed to cause any increases in the amplitude of vibration, and if the rubbing was not too severe the shaft would clear itself.

It is, of course, not to be assumed that rubbing contact will in all cases start along that element of the shaft which should run out, according to the simple theory of shaft behavior outlined above. The shaft might be out

† Another rather violent effect was found to follow when this model was subjected accidentally to *severe* rubbing. The rubbing surface was a wooden bushing in a metal guard ring surrounding the shaft with considerable clearance, and carried on a rigid standard. Upon several occasions when the rubbing became too severe the tangential force of friction caused the shaft to roll around on the inner surface of this wooden ring with great violence. The frictional resistance to rotation was so great that the model was brought to rest quickly. This phenomenon does not occur unless the shaft rubs very hard on a solid ring rigidly held.

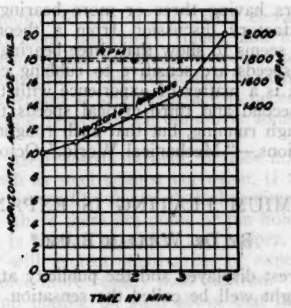


FIGURE 6.

of round, or bowed and balanced in such a manner that some other element becomes heated when rubbing begins. In any case, however, light rubbing will almost certainly heat the shaft unevenly, and that element of the shaft which gets the most heat will tend to expand and so to bow the shaft. This will change the unbalance of the rotor in the manner of the experiment made, in which a radial bolt was used to bow the shaft (Figure 5). The effect of rubbing, then, is to tend toward making the rubbed side the heavy side, and consequently to bring on the vicious cycle described above if the rotor runs below its critical speed, and to relieve the rubbing, or, at least, not to build it up, if the rotor is running above its critical speed.

Consider, for example, two hypothetical cases, in both of which it will be assumed that the rubbing starts on the side of the shaft exactly opposite to that indicated by theory.

Suppose, first, that with a shaft running *below* its critical speed the rubbing starts on the side of the shaft opposite to the location of the unbalance, due to a ridge or prominence of the shaft on that side. The *rubbing* should increase in violence, even though it improves the *balance* at first by bowing the shaft outward on the rubbed side. The bowing outward on the rubbed side increases the severity of the rubbing. Very little bowing of a shaft (say, five-thousandths of an inch at the center) is sufficient to produce a predominant unbalance on the bowed side. The rotor would soon get badly unbalanced and the cycle should proceed as described above.

In the second case, suppose that a shaft running *above* its critical speed rubs on the side of the unbalance, due to some ridge or other prominence of the shaft on that side. The unbalance would increase on account of this rubbing, but owing to the fact that the heavy side runs in when a rotor runs above its critical speed, the shaft would simply bend backward as described above (illustrated in Figure 5), so as to withdraw the prominent element from the rubbing surface. In this case, therefore, the vicious cycle would not occur.

It is by no means intended to give the impression that rotors running below their critical speeds cannot endure any rubbing whatever. The fact is that such rotors do frequently rub and clear themselves. They are, however, sensitive in this respect. The object of this note is to call attention to the relative freedom from such sensitiveness of rotors which are running above their critical speeds.

The cases of rotors having three or more bearings are too complex to admit of very satisfactory discussion from a theoretical point of view. However, experience seems to show that three-bearing rotors running below their lowest critical speeds are sensitive to rubbing contact with the shaft. On the other hand, it is a matter of experience with three-bearing sets running between their second and third critical speeds that rubbing the shaft sometimes causes rough running, but that such roughness does not build up to dangerous proportions.—"Mechanical World," October 22, 1926.

## CHROMIUM PLATING IS EXPANDING.

By Dr. William Blum.*

Judged by the interest displayed and the publicity attached to this subject, chromium plating might well be called the sensation of the last few years in the field of protective coatings, a field in which there have been but few other radical changes in the last decade. Among questions that are no doubt now in your minds, and in those of many chemists and engineers, are "What is the future of chromium plating?" "Is it a practical process?" "What is its probable value when applied to engineering materials?" "In short, can we use it?"

In this informal talk I will attempt to answer, or at least to discuss, some of these questions, especially from the standpoint of the mechanical engineer. In doing so I am limited principally to the experience and contact of the Bureau of Standards, because so little authentic information has been published on this subject, especially in this country. As, however, a large number of industrial research laboratories are now engaged in the study of chromium plating, and hundreds of firms are trying it out for various purposes, it will no doubt be possible within a few years to make much more definite statements than are now justified.

### PROCESS HAS SOME DEFINITE ADVANTAGES.

In order for any new process or product to compete permanently with the established procedure, it must have certain definite advantages. It may, therefore, be well, before discussing methods and applications of chromium plating, to consider those properties of the metal which make it of special interest.

The outstanding property of chromium, when deposited under appropriate conditions, is its extreme hardness, as measured, for example, by its resistance to scratching. A bright chromium deposit, when tested in the Bierbaum apparatus with a sapphire point and a given load, yielded a scratch with a width of about 0.7 micron (the narrowest scratch of any metal thus far examined), while the cold-rolled steel on which it was deposited yielded a scratch about 2.2 microns in width. The great hardness of the chromium, and the fact that this as well as its other properties may be varied by the proper choice of plating conditions, at least justifies its consideration wherever hardness is an essential factor.

Another property of chromium that is distinctive is its resistance to tarnish or oxidation under many conditions of exposure. Thus it will stay bright for long periods, not only in an ordinary atmosphere, but also when

* Chemist, United States Bureau of Standards, Washington, and president of the American Electrochemical Society. Contributed by the Machine Shop Practice Division for presentation at the annual meeting, New York, Dec. 6 to 9, of the American Society of Mechanical Engineers.

exposed to a high humidity, to salt air, to fairly elevated temperatures, to molten tin and zinc, to many laboratory fumes, and to concentrated nitric acid. It is readily attacked and dissolved, however, by hydrochloric acid, and more slowly by sulphuric. This resistance to tarnish, therefore, justifies its consideration wherever a bright surface is necessary, as on mirrors, even though the reflecting power of chromium is somewhat less than that of silver.

Although chromium itself resists tarnish, it does not necessarily protect an underlying metal such as steel against corrosion, if the steel is anywhere exposed. Thus it may be readily shown that chromium-plated steel quickly corrodes at any points where there are pores or pin holes in the deposit. In this respect chromium is similar to nickel and copper, and unlike zinc and cadmium, which latter will protect small areas of exposed iron because the zinc and cadmium dissolve more readily than the iron. Any superior protective action of chromium plating on steel above that of nickel plating must depend upon producing more nearly impervious deposits of chromium than of nickel. From present indications it appears probable that for such uses chromium will generally be applied over nickel plating of good quality, in which case the chromium is chiefly useful for its hardness and tarnish resistance.

With the definite advantages it possesses, it is no doubt surprising that chromium plating has been such a recent development. This condition is not the result of any lack of interest in the subject, but from inability until recently to so define and control conditions that consistent results may be obtained under industrial conditions. Chromium was electrodeposited by Bunsen as early as 1854, and since then numerous papers and patents have been issued on this subject. Of the more recent papers, that published by Sargent (†) in 1920 is probably of greatest interest, as most of the methods proposed since that date represent modifications of Sargent's solution, or of his operating conditions.

Sargent recommended a bath of which the major constituent is chromic acid ($CrO_3$) in a concentration of about 2.5 M or 250 g. per L. (33 ounces per gallon). To this he added a small amount, 3 to 5 g. per L. (0.4 to 0.7 ounces per gallon) of chromium sulphate, $Cr_2(SO_4)_3$. From such a bath, Sargent and others have obtained good chromium deposits, but frequently the results have been erratic. H. E. Haring, from a study at the Bureau of Standards ("Chem. and Met. Eng.," vol. 32, page 692, 1925) concluded that in such a bath it was necessary to regulate the acidity. This is accomplished practically by having present in the bath a colloidal suspension of chromium chromate, which may form automatically or may be produced by the addition of any basic or reducing substance.

### DEPOSITS ARE OF THREE TYPES.

Even more important than the exact composition of such a bath is the control of operating conditions, especially the temperature and current density. These have an appreciable effect in all plating operations but in chromium plating relatively small variations in these factors may change entirely the character of the deposit, or even prevent deposition entirely. Three principal types of chromium deposit may be produced, though of course these shade gradually into each other:

(1) At too low a current density or too high a temperature, a "milky" deposit is produced. This is relatively thin owing to the very low cathode efficiency under such conditions.

(†) *Trans.* Am. Electrochem. Soc., vol. 37, page 49, 1920.

(2) At the appropriate temperature and current density, e.g. at 45 degrees C. (113 degrees F.) and 10 to 20 amperes per square diameter (100 to 200 amperes per square foot), a bright deposit is produced.

(3) At too high a current density or too low a temperature, the deposit becomes "frosty," gray and "burnt."

Of these deposits, the milky form is the softest, and the bright is the hardest, as measured by the scratch test.

With an appropriate solution, temperature and current density, it is a relatively simple matter to produce bright, hard deposits of chromium upon those articles where a nearly uniform current density may be secured, for example, upon nearly plane surfaces, cylinders, etc. Upon irregularly shaped articles, and especially those having deep recesses, it is very difficult to get a continuous deposit of chromium of uniform properties. Thus it may then be found that no metal is deposited in the recess, or else the deposit on the projecting parts is gray and spongy. The latter type of coating is hard to buff to a bright surface.

This poor throwing power of the chromium solution is due principally to the fact that the current efficiency decreases rapidly as the current density is lowered. Thus at 45 degrees C. (113 degrees F.), the cathode efficiency at 20 amperes per square diameter 200 amperes per square foot) is about 18 per cent, while at 5 amperes per square diameter (50 amperes per square foot) it is only about 7 per cent, and at slightly lower current densities it is practically zero. This limitation appears to be an inherent defect of baths containing chromic acid, and while minor improvements in the throwing power may be effected, there is little reason to believe that it can ever be made to approach that of a nickel plating bath, much less that of a cyanide copper solution.

Another factor which may affect the introduction of chromium plating on a very large scale is the relatively large power cost involved. Ordinarily in electro-plating the cost of power is negligible, or of the order of 1 cent or less per square foot. With chromium, however, the power cost may be from five to ten times as great. This is due to several factors, especially (1) the low electro-chemical equivalent of chromium in chromic acid, in which it has a valence of six; (2) the low cathode efficiency of chromium deposition, generally about 15 per cent, and (3) the higher voltage, generally 8 to 10 volts, required by the use of insoluble lead anodes and high current densities.

### COST OF CHROMIUM IS NOT PROHIBITIVE.

The cost of the chromium itself is not prohibitive, as the metallic chromium in chromic acid at 40 cents per pound costs about 80 cents per pound as compared with nickel at 45 cents. A coating as thick as 0.025 millimeter (0.001 inch) represents only about 3 cents worth of chromium per square foot. The total cost of chromium plating is certain to be somewhat greater than that of nickel, as the investment, the power, and the labor cost are all higher in chromium plating. On those products for which the chromium has unique advantages, and especially where the general labor cost represents a large part of the entire expense, the greater cost of the chromium plating may be fully justified.

Among the possible applications of chromium plating, those dependent upon its great hardness are especially promising. One interesting example is its use on plates for printing currency and securities at the United States Bureau of Engraving and Printing, where the process and conditions developed by H. E. Haring at the Bureau of Standards are in successful operation. Some years ago the Bureau of Standards cooperated in the

design and installation of a plant for reproducing these plates electrolytically. The plates produced by this process have a nickel surface, followed by alternate layers of copper and nickel, finally sweated to a steel plate. These plates are "intaglio," that is, the designs are below the plane surface, and before each impression the plate is inked and rubbed first with burlap and then with the hand. As all inks contain some abrasive particles, there is considerable wear on the plates. As was expected, the plates with a nickel surface did not last so long as the case-hardened steel plates that were formerly used exclusively.

By the application of about 0.005 millimeters (0.0002 inch) of chromium to the electrolytic nickel plates, it was found that their useful life may be increased to about four times that of the nickel or twice that of the hardened steel plates. The impressions are usually sharper, and through the use of a smaller number of plates, greater uniformity and security of the product is obtained.

In other branches of the printing industry, it has been found that when very long editions are required, as of labels and soap wrappers, chromium-plated electrotypes may be used to produce from three to six times as many impressions as can be made from the nickel electrotypes.

### CHROMIUM INCREASES LIFE OF GAUGES.

Of more direct interest to engineers is the application of chromium to gauges. In a recent study at the Bureau of Standards (‡) the performance of chromium-plated plug gauges was compared with that of hardened steel gauges. In these comparisons a wear testing machine, devised by the above authors, was employed. By its use, two gauges were automatically moved up and down in hardened steel rings, and the wear was measured after a determined number of such gaugings. From this study the following tentative conclusions were reached: When exposed to sliding friction, with no abrasive present, the chromium-plated gauges resisted wear about five times as well as any of the steels tested. When, however, abrasives such as fine emery were present, the chromium-plated surface, while still superior to the steel, was only 30 to 60 per cent better. This latter result does not, however, mean that the chromium plating is unsuitable for resisting wear by finely divided abrasives such as emery under all conditions. Thus it was found in lapping wear tests that the chromium resists wear from two to four times as well as the customary gauge steels. It is at least probable that, by depositing the chromium under different conditions, coatings may be produced which are best suited to resist each particular type of wear. For all those conditions of service in which chromium-plated gauges may be found applicable, they have the advantage that a relatively soft steel may be used as a base and thus dimensional changes with time may be avoided.

In the experiments thus far conducted at the bureau, a relatively thick chromium deposit (about 0.02 millimeter or 0.0008 inch) was applied, after which the surface was ground and lapped to the desired dimensions, leaving a somewhat thinner coating of chromium on the gauge when it was tested. A simpler and more economical procedure would be to apply a relatively thin chromium layer (e.g. 0.005 millimeter or 0.0002 inch) to a finished, accurately under-dimensioned gauge. The latter could then be used directly after plating and, after a pre-determined length of service during which about half the chromium would have been removed, the remaining chromium could be readily dissolved off and a new coating applied. Whether such a

(††) H. J. French and H. K. Herschman. Preprint 18 of the American Society for Steel Treating, September, 1926.

procedure will prove practicable remains to be seen. While both laboratory tests and plant practice have indicated that chromium plating may not be universally advantageous on gauges, the results are certainly sufficiently promising to warrant further investigation and trial.

The experience with gauges at once suggests the application of chromium to other steel surfaces exposed to wear, and which now require special hardening processes. A few observations indicate that chromium plating on certain cams is practical and advantageous. On gears, it may be difficult to produce satisfactory deposits in the depressions, and the wear on the teeth is much more likely to detach the chromium coating. On stamping dies or other surfaces exposed to severe impact, it is at least probable that a light chromium coating would furnish little protection against the deformation of a soft steel base. If, however, chromium can be made to adhere permanently to a case-hardened die, it would preserve the details. On dies used in molding plastic materials, the application of chromium will probably be advantageous. More extensive experience than is now available, or at least published, will be required before a final conclusion can be reached regarding these and similar possible applications.

Uses of chromium dependent upon its resistance to some specific corrosive condition include such applications as the following: Molds for vulcanizing rubber may be chromium-plated to prevent sticking of the rubber to the mold. The resistance of the chromium to sulphur or many sulphur compounds, which leads to the above use, has also caused its consideration on oil-cracking equipment. The resistance of chromium to oxidation has suggested its use on glass molds, and on rollers for making plate glass. So far as is known, the latter two uses are still in the experimental stage.

### CHROMIUM PLATE MAKES GOOD REFLECTORS.

The luster and permanence of chromium plating warrant its consideration for reflectors, especially those that are exposed to sulphur fumes such as in locomotive headlights and flood lights. Even though its reflective power is only about 60 per cent as compared with 90 per cent for silver, the rapid tarnishing of the silver more than compensates for this initial difference.

Whenever a bright surface is required, that is not exposed to severe corroding conditions, the application of chromium directly to steel will be advantageous, for example, on rules and scales for shop use. Where, however, steel articles are to be exposed to the weather, it will be found desirable to apply a coating of chromium (0.005 millimeter or 0.0002 inch) over a substantial nickel coating (of 0.025 millimeter or 0.001 inch) or still better over a coating composed of a copper and a nickel layer, or a nickel, copper and nickel layer. In the automobile industry great interest is being shown in such applications of chromium plating, and one make of car is now made with chromium plating on the radiator. As previously indicated, the value of chromium under such conditions depends largely on its luster, and its resistance to tarnish and abrasion. There is little reason to believe that the chromium will materially increase the resistance to corrosion of the steel under severe conditions of exposure.

On brass articles, such as plumbing fixtures, where there is little tendency for the base metal to corrode, the chromium may be plated either directly on the brass, or on a nickel-plated surface. In the latter case, however, it is essential that the nickel plating should be very adherent, otherwise it will peel during the chromium plating. The more rapid adoption, or at least trial, of chromium plating for such fixtures is undoubtedly hampered by

the poor throwing power and the greater personal attention required for consistent production.

With so many research laboratories engaged in the study and development of chromium plating, it seems safe to predict that even though no revolutionary developments appear probable, at least with the present type of bath, a fund of experience will soon be gained, upon which will be based the application of chromium plating to those many purposes for which it is especially suited. Chromium plating is not a panacea; it will not replace all other forms of plating. It will serve, and indeed has already served, many purposes better than other metals, and some that other metals cannot serve. —"The Iron Age," December 16, 1926.

## PROGRESS OF MARINE ENGINEERING.*

By Engineer Vice-Admiral Sir Robert B. Dixon, K.C.B.

Engineer Vice-Admiral Sir Henry Oram, one of my distinguished predecessors, in his presidential address to the Institution in 1909, reviewed the progress made in naval engineering to that date and indicated some possibilities then in sight, while holders of the office in later years have also brought to notice current developments and changes in marine engineering practice.

I am told that this subject continues to hold your interest, and I purpose therefore to continue the narrative of marine engineering progress with particular reference to naval practice.

### CONSIDERATIONS OF NAVAL DESIGN.

It is desirable to refer in the first place, in order to promote an understanding of the special problems, to the special conditions by which naval design is circumscribed and to show why certain changes which permit advantages in other machinery are denied to the Navy.

To a considerable extent the requirements for naval and commercial machinery run on parallel lines in so far as reliability and durability are the outstanding requirements. These characteristics must be kept well in the foreground. While economy in fuel is a valuable and desirable quality, it has to give way to a varying extent in naval designs to saving in weight, to the need for simplicity, to accommodation in a limited space both in respect of floor space and height, and of the subdivision of the machinery in a number of separate water-tight compartments. These considerations sometimes combine to outweigh the matter of running costs which is necessarily a paramount consideration in commercial work.

A measure of the progress attained in ten years is indicated broadly by the comparsion of H.M.S. *Lion*, referred to in my predecessor's address, and H.M.S. *Hood*. The comparison marks the transition from coal to oil, from direct-driven turbines to geared turbines, and from the large-tube water-tube boiler to the small-tube water-tube boiler.

The horsepower of *Hood*, both in the aggregate and unit, is double that of *Lion*, and is provided for on an equal floor space and on an equal machinery weight. The engine-room complement in *Hood* is one-half that of the *Lion*. Thus, in ten years, each ton of machinery weight and each square foot of machinery space were made to do double duty, while each man replaced four who were necessary to operate the earlier design.

* The Junior Institution of Engineers. Presidential address delivered in London, December 3rd, 1926. Abridged.

The reduction in weight, space and complement is in the main attributable to the change from coal to oil, but the small-tube boiler and the large boiler units possible with this type also contribute to the advance, as does also in respect to space the geared turbine, observing also that the geared design contributes to the attainment of a higher ship speed on the given horsepower by virtue of the improved propulsive efficiency which attends the lower speed of revolution of the propellers.

The influence of the type of fuel and the large boiler units is particularly striking in the case of the torpedo-boat destroyer class of vessel, in which are fitted boiler units having a heating surface of 8000 square feet and evaporating water at the rate of 18 pounds per square foot of heating surface per hour. Here installations of 27,000 shaft horsepower are produced for a total machinery weight including water of 33 pounds per shaft horsepower, and they operate at sea with a total engine-room complement of twenty-eight.

The advantages of oil fuel are generally well appreciated and need not here be recounted. It is not so well realized, however, that in addition to the direct advantages such as saving in weight and man power, in economy and in convenience which attends its use, it has alone permitted of the development of boiler units in sizes quite impracticable with hand-fired boilers, and so it has exercised a secondary favorable reaction on the design problem. Boiler units of 10,000 shaft horsepower are common naval practice today, and they contribute in an important degree to the saving in weight, space and complement to which reference has been made.

### MECHANICAL GEARING.

Although the direct-driven turbine designs of lower power had given very little trouble on service, the prospect of very large units was viewed with some apprehension in view of the possibility of distortion due to irregular expansion and the effect thereof on the reliability, having in view the relatively fine turbine clearances, which economy in performance entails. These facts, coupled with consideration for the propeller efficiency, induced the Admiralty to follow with close interest Sir Charles Parsons' early experiments on mechanical gearing in mercantile designs, and to arrange for an early trial of the system. After experience in two torpedo-boat destroyers of the partially geared type, two torpedo-boat destroyers of the 1913 program were arranged to be fitted with all-geared turbines, and these vessels came on service at the outbreak of war after satisfactory trials. The trials fully confirmed the promise in improvement, showing in comparison with sister vessels an improved speed and an increase in distance steamed per ton of fuel of 14 per cent at full speed with the boilers forced at lesser rates. An even more striking gain was revealed at cruising speeds where the distance steamed per ton of fuel increased by some 70 per cent, this being due in part to the use of cruising stages to which the all-geared designs lend themselves more readily than in the larger direct-driven designs. The machinery weight was sensibly the same as in the sister vessels referred to, the extra weight entailed by the turbines and gearing being set off by a saving in boiler weight which the improved turbine consumption permits.

Quickened by the greatly increased construction entailed by the war, the application of the geared design to all classes of ships was rapid, and by the end of the war installations aggregating five million shaft horsepower were on service afloat, ranging from the P. boat of 3500 horsepower to the cruisers of 90,000 horsepower and including steam-propelled submarines. The fact that the performance under the severe conditions of warfare was

entirely satisfactory and gearing defects were few, is a tribute to the soundness of the principle, especially when it is considered that gear cutting on this large scale is a relatively new art.

The use of mechanical gearing has fulfilled essential purposes in the development of naval machinery. It has permitted a reversion to the propeller efficiencies which were necessarily sacrificed in varying degree on the occasion of the introduction of the turbine, while occasioning no net increase in machinery weight. The case of the torpedo-boat destroyer class may be quoted in which this figure was increased from 40 per cent in the worst cases with very fast-running direct-driven propellers to about 60 per cent. It has yielded an improvement in fuel economy and particularly so at cruising speeds, thereby the steaming endurance has been sensibly increased.

It has permitted of the considerable augmentation of power per shaft entailed by the higher speeds demanded in large ships while employing turbines of reasonable size; thereby the reliability is improved and the problems arising in the use of increased steam temperatures and pressures have been simplified.

Over and above these advantages, it permits, without otherwise prejudicing the design, of self-contained units of power, observing that in the direct-driven designs for the larger ships the turbines on the respective propeller shafts are in varying degrees interdependent in so far as they are necessarily compounded to ensure the length of steam path necessary to attain the desired efficiency. To this extent the direct-driven designs suffer in respect of vulnerability by comparison with the geared, as in the latter case damage to any one turbine affects only the performance of its own propeller shaft.

### ALTERNATIVE SYSTEMS OF POWER TRANSMISSION.

The alternative systems of transmission of power by hydraulic and electrical means have not been favorably regarded in this country. The former system, in the shape of the Fottinger transmission gear, developed in Germany just prior to the war, was not extensively used, but a somewhat similar device has been introduced recently for use in merchant ships in association with geared oil engines, this being, however, a flexible transmission device rather than a means of reducing the engine speed, which also permits of the engine running with the ship at rest.

The electrical transmission system has only been fitted in isolated installations for mercantile work in this country, in association with oil engines in one design, but it has found considerable use in the U. S. A., where it is employed in all battleships of the U. S. Navy completed since the closing years of the war. This important development has been the subject of considerable discussion and its relative merits in comparison with mechanical gearing have been fully ventilated in the technical press and elsewhere. There is no doubt that the electrical systm as fitted in the United States ships is a proved engineering proposition and that it has given satisfactory service under peace conditions. In comparison with mechanically geared installations the design entails increased machinery weight, and under war conditions it is considered likely to be less economical in fuel and exposed to greater risks of breakdown or disablement. On these grounds, and notwithstanding advantages in some directions, the system has been considered on balance to be less suited to our Naval purpose than mechanical gearing, observing further that in any case mechanical gearing has been found to be necessary in the lighter classes of war vessel in all the principal navies to meet the more severe restrictions in machinery weight which operate in these designs.

An electrical transmission system in association with heavy oil engines as the source of power has, however, been fitted in H.M.S. *Adventure* for use at cruising speeds for the purpose of obtaining a large radius of action at low speeds. This plant is separate from the main steam-driven propelling machinery and will not be in use at speeds higher than cruising speeds.

### SUPERHEATED STEAM.

The naval problem in respect of superheat differs in other respects from the commercial or mercantile marine problem. The latter installations may be expected to operate for the greater part of their time at a high proportion of full load and accordingly fuel economy at low outputs is of little importance. The naval vessel, however, steams for the greater proportion of its service at cruising speed, and while economy in fuel at high speed is essential, it is also necessary that good economy shall be realized at cruising speeds in order that fuel may be conserved and a large steaming endurance obtained on a given fuel stowage. Under cruising conditions only a portion of the boiler installation is in use, and on grounds of reduced wear and tear and fuel economy the plant in use is operated at a relatively low output. Accordingly it is necessary that the design of superheater should be such that a reasonable degree of superheat will be attained under these low power boiler working conditions. Thereby the additional weight and complication attending the use of superheaters is used to the greatest advantage.

Separately fired superheaters are out of the question on account of the additional weight and space that would be entailed. The usual designs of superheaters in use at that time provided in general for the increased temperature being given to the steam at the position where the funnel gases left the generator system, and in such designs the superheat attained at low outputs of the boiler is inappreciable. In any case, and especially in the naval designs where the further exploitation of the waste heat in the funnel gases by feed or air heating has hitherto been rendered impracticable by the design restrictions, such a system, while convenient in many respects, is open to objection on fundamental design principles, in that a given weight of material is not used to the best advantage.

On the other hand, the placing of the superheater element in its logical position in the gas circuit is not without its difficulties, and experimental work was necessary to determine how the conflicting requirements, namely, a remunerative degree of superheat at low outputs and a safe and durable design at full outputs could best be met.

It has been found practicable to devise a design in which as much as 70 per cent of the maximum superheat is realized with the boiler working at one-fifth output. Exhaustive shop trials of boilers of this type providing an evaporation sufficient for 10,000 shaft horsepower have been carried out with satisfactory results and boiler efficiencies of 80 per cent and 72 per cent respectively have been obtained when burning fuel at rates of 0.2 pound and 1.0 pound of oil fuel per square foot of total heating surface per hour. The boilers referred to are the usual three-drum small-tube type as modified for the application of superheat, which, as already mentioned, are not associated with any devices for tapping the waste heat in the funnel gases after leaving the generator system. Some difficulties, not altogether unanticipated, have arisen in regard to the materials employed for the superheater supports and fittings external to the pressure system, but they have been surmounted by the employment of the special heat-resisting materials which,

thanks to the efforts of the metallurgists, are now available or in course of development to meet the many increased requirements to hand or in sight for material possessing improved qualities in respect of maintaining their rigidity under high temperature conditions.

## WATER-TUBE BOILERS.

The commercial demand for larger boiler units and for increased output, which considerations are combining to effect an increasing use of the water-tube boiler, both on shore and afloat, suggests that some observations on naval experience with such boilers may be worth putting on record. It is now over thirty years ago that the use of the water-tube boiler became general for all classes of warships. Many types have been used in the interim, as is generally known, but in recent years the three-drum type with small tubes of from 1⅛ inches to 1¼ inches external diameter has been exclusively employed, this type being best suited to meet the needs for the relatively large boiler units and reduced machinery weights entailed by modern warship developments.

It need hardly be said that the quality of the tubes is a matter of first importance, and considerable attention has accordingly been given to the processes of manufacture to ensure the desired finish and quality of the seamless steel tubes used for naval water-tube boilers. At one time foreign steels were considered necessary by the manufacturers to produce tubes to meet the Admiralty requirements and tests, but it is gratifying to record that during recent years the desired results have been obtained from British produced steel.

The specification provides for seamless tubes to be finished by a cold drawing process to ensure a smooth surface inside and outside, and importance is attached to the observance of this condition from the aspect of the improved durability of the tubes which experience shows is thereby attained. Importance, too, is attached to the annealing of the tubes after the drawing process and also after any work involving bending during the boiler shop processes.

The mechanical tests, which include tensile, temper, flattening, bell-mouthing and expanding tests, are carried out at the tube makers' works before acceptance. After receipt at the boilermakers' works the tubes are electro-zinced externally, and this process affords a means of ensuring entire freedom from surface defects as well as a protection against corrosion during the erection of the plant. Finally, each tube is water tested internally to a pressure of 2500 pounds per square inch.

These tests, which are not, of course, all carried out in the case of each individual tube, a parcel test meeting the case in several instances, have proved to be necessary to ensure against any failure on service from the purely material aspect. It is to be observed that in fitting in place in the tube plate each tube is subjected to the rolling and bell-mouthing process, and this affords a useful check on the suitability of every tube in respect of structure and ductility.

Given pure feed and proper use of the feed-heating appliances, the life of boiler tubes of the small-tube water-tube boiler, which range in thickness from .104 inch to .128 inch, is found to average about ten years under peace-time conditions.

It is the practice to carry out periodical drill and wear and waste tests, the interval between which is reduced as the age of the boiler increases. Sample tubes are cut out and split for examination and gauging, and by these means the useful life of the boiler tubes can be predicted very closely.

to the scattered disposition of the auxiliaries and the widely varying requirements. There is, too, the question of feed heating; the use of the regenerative feed heating system is not practicable, mainly owing to the complication entailed by the provision required to make proper use of it at all rates of power of the main engines. Feed heating is essential under all conditions of working, both for economy and for durability, and this can in the special circumstances be most economically attained by continuing the use of steam in the case of certain auxiliaries and employing their exhaust steam for feed heating.

The use of higher steam temperatures is likely to lead to the decreasing use of the steam reciprocating engine, in so far as the internal lubrication required leads to the necessity of greater precautions in filtering the feed and to the possibility of greater risk of damage in the highly forced boilers which are liable to damage unless the feed is free from grease or oil. Thus the tendency is to employ turbine-driven auxiliaries in the cases where they are to be driven by steam, except in the case of the standby auxiliaries, which may be electrically driven. In general, the electrically driven auxiliary entails rather greater weight and space. In comparison with the slow-running reciprocating engine the fast-running motor-driven auxiliary is, of course, lighter, but, when allowance is made for the extra provision which has to be made in the form of electric generating power to provide the current, the net result is as stated. The concomitant advantages, however, which include the improved economy referred to, freedom from vibration—an important consideration in warships, where the delicate apparatus necessary to insure the full use of the fighting appliances have to be safeguarded in this respect—and the absence of steam pipes to outlying spaces make the change on the whole a profitable one.

### OIL ENGINES.

Considerable progress has been attained during the period under review in the application of the heavy oil engine for use in marine propulsion. Its suitability for the slower types of mercantile vessels in which steam held, a few years back, indisputable sway has been established definitely, and for such designs the balance of advantage as between oil engines and steam appears now to be determined by financial considerations rather than on purely technical grounds, keeping in view the intended service and itinerary of the ship. This rate of progress affords, I think, a remarkable testimony to the skill and ingenuity of the workers concerned in the development who have faced and overcome great difficulties.

In its application to faster designs entailing more powerful engines the issue is not so clear cut, and the technical issue of the development of higher powers in a single cylinder enters into the question as well as the question of first cost. These two aspects are closely related one to another and to the problem as it presents itself to the naval designer; that is, the improvement of the power engine-weight ratio, which is at present, after making due allowance for the improvement in fuel consumption, much inferior to the oil-fired boiler—steam turbine combination.

The further prospects of the oil engine are viewed with considerable enthusiasm in many quarters, but I believe that this optimism is not entirely shared by those in close touch with the technical problems.

In considering the matter of its wider and general application in large units of power in other directions, as, for example, electric power generation or marine propulsion, regard must also be paid to the progress of steam plants. Impelled by the commercial necessity for cheap power and

possibly spurred by the high standard in fuel economy set up by the heavy oil engine, the fuel economy of the steam plant has made very great advances in recent years, and in this respect its performance is converging towards that of the oil engine. At the present day a standard of economy represented by the generation of one kilowatt of electrical energy from 1 pound of coal per hour has been fully demonstrated as commercially attainable in power plants of any output, however large, within the range of modern requirements. That standard, translated to its equivalent marine basis, corresponds to the consumption of a little more than half a pound of fuel oil per shaft horsepower per hour and is susceptible of some further improvement by the use of higher steam pressures. Further, there is promise of sensible improvement if the use of some binary fluid system is found to be commercially practicable.

Admittedly, the complications attending the realization of economies of this order are considerable and greatly in excess of that of the steam plants hitherto used and on which the reputation of the steam plant for flexibility, durability and reliability of operation has been established. On the grounds of complication, and possibly also for reasons of weight, space and/or first cost, the complete application of the steam developments to marine use does not present itself at present as being a practicable or prudent course. Yet considerable use may nevertheless be made of them, and there is no reason why, as confidence in the advanced types is gained, a marked advance in fuel economy beyond that at present obtained should not be realized in steam plants afloat.

Turning again to the oil engine, there is no question that in respect of fuel economy it has promised and attained a standard considerably in excess of that given by contemporary steam plants; but it appears that some of the disabilities from which it suffers in an increasing degree as its size and power output are increased may possibly be somewhat obscured by this outstanding advantage. I have referred in passing to the problems attending its further development as regards higher unit powers and cheaper and lighter manufacture, and without attempting to cover the whole ground I would invite your close attention to certain of them from the designers point of view. In comparison with a prime mover, such as a steam turbine, the design of a heavy oil engine presents much greater difficulties. The steady uniform stresses and temperatures in the structural members of a steam plant in operation are replaced by rapidly fluctuating stresses and temperatures. The mechanical stresses to which the material is exposed are, at certain parts of the cycle, out of all proportion to the mean load transmitted, and these stresses are in the case of certain components augmented by heat stresses arising from temperature variations, the higher temperatures being, moreover, of such an order as to prejudice the resistance of the material to meet these stresses. The rubbing parts within the cylinder, on whose efficiency depends the efficiency of the cycle, are exposed to very high temperatures at which lubrication, in the ordinarily understood sense of the word, becomes doubtful. The combustion problem, which embraces the design of the exact mechanism required to give precision in point of time and quantity for the fuel injection, no less than the complete devices to provide for the pulverization and dissemination of the liquid fuel in close association with its air quantum throughout the combustion chamber, on which its complete combustion in an exceedingly short space of time depends, becomes evidently of greater complexity as size and speed of operation advance.

All the difficulties mentioned increase in varying degrees as the engine power is increased, whether the advance be attained by increase of size,

revolutions or increased mean pressure or by combination thereof, and in the face of them the rate of progress towards the production of dependable units of large power and lighter and cheaper engines appears likely to be slow. Indications, moreover, point to a stage in the development of the larger oil engine being reached, when the auxiliary power required to supply the air for combustion will assume such proportions that the high efficiency of the cycle will be discounted appreciably, and, while means may finally be developed for countering this falling-off by retrieving some of the energy now wasted in the exhaust gases, any appliances of that type will increase the first cost and complication to an appreciable extent.

### CONCLUSIONS.

I have endeavored in my remarks to convey some idea of the progress attained, and, without venturing to prophesy as to the expectations for the future, to indicate some of the problems that have to be faced in seeking further progress. Actually, as may be inferred from my earlier remarks, the same basic problems really arise whether the developments are intended for naval or commercial work. In short, what is required, while maintaining a proper standard of reliability, is improved fuel economy and improved durability, and that these improvements shall be realized on a lesser machinery weight and on lesser first cost. The designer's underlying aim is accordingly much the same in either case, and, expressed in a few words, is the application of the material and of the human effort employed in the construction of the machine to the fullest advantage.

Nature is chary of disclosing her secrets, and it is evident that each further step will lead to problems of still greater complexity, and, judged by past standards, out of all proportion to the degree of advance contemplated. Hence it follows that the stages of development in future will be more strongly marked by the need for close specialization, and the net of the designer must be cast even wider to gather for his needs the specialist work of investigators outside the immediate sphere of the practical engineer.

While acknowledging, therefore, with gratitude the debt we, as engineers, owe to the mathematician, physicist, chemist, metallurgist and kindred scientific workers, and to the numerous specialist workers in the wide field of modern engineering practice for the advances which their work has alone permitted us to make, I suggest we should keep well in mind the definition of the term gratitude in the more subtle sense, namely, a lively anticipation of future favors which the growing complexity of our art will need to be conferred on us if progress is to be maintained.

In conclusion, I desire to emphasize the necessity for co-operation and co-ordination of effort, not only in the organization of each individual engineering concern, but in the industry as a whole, in order that satisfactory progress may continue. Much has been done in this direction in recent years by the engineering institutions and other bodies, which stand essentially for united effort and the wider dissemination of information, but much remains to be done to avoid the quite unnecessary duplication of effort and overlapping of research work which exists today. In particular, the pooling of technical data and experiences of individual commercial undertakings to a much greater extent than in the past is greatly to be commended, and if this can be attained in an increasing degree it can hardly fail to lead to speedier and more certain progress in the elucidation of the increasingly difficult problems with which we are confronted.—"The Engineer," December 3, 1926.

## AERO-ENGINE VALVE STEELS.

As was the case with many problems connected with the working of high-duty internal combustion engines, the question of the selection of a metal for valves, capable of withstanding the exacting conditions imposed, came into prominence during the European War. Under the leadership of the Director of Research at the Air Ministry, exhaustive tests on the subject of valve steels were carried out at the National Physical Laboratory and at various other research centers. A very able summary of the work accomplished was embodied in the form of a paper read before the Institution of Automobile Engineers, by Dr. L. Aitchison, in November, 1919. Although this pioneer contribution to the study of the valve-steel question is now seven years old, it still remains in many respects a standard work. The general conclusions arrived at by Dr. Aitchison were that a 3 per cent nickel steel was suitable for all inlet valves, and for exhaust valves working at a temperature not above 600 degrees C. For exhaust valves, subjected to temperatures of between 600 and 760 degrees C., a high-chromium steel was suggested, and for temperatures above 760 degrees C., a tungsten steel was favored. As far as motor-car engines are concerned, the use of plain 3 per cent nickel steel is now almost universal, and the material has given consistently satisfactory results. In the case of valves operating at high temperatures, however, opinions are by no means unanimous, and the conclusions put forward from time to time by various workers in this field differ considerably. In a paper read before the annual meeting of the American Society for Testing Materials in June, 1924, Messrs. J. B. Johnson and S. A. Christiansen found that for operating temperatures which might reach 815 degrees C., the quaternary alloys of iron, chromium, nickel, and silicon, in which the chromium ranges from 6 to 15 per cent and the nickel from 20 to 25 per cent could not be used, since they could not be hardened sufficiently to prevent excessive galling of the stems and wear on the tips. Again, steels containing from 15 to 18 per cent of tungsten and from 3 to 4 per cent of chromium, retained, as would be expected, their original hardness, but scaled heavily at the valve seat. The material which gave the best results in Messrs. Johnson and Christiansen's experiments, was a steel containing from 11 to 13 per cent of chromium and from 3 to 3.5 per cent of cobalt, together with a little molybdenum. This steel, although not completely satisfactory, was recommended for ordinary operating conditions at high temperatures.

It is interesting to compare the findings of these two workers with the results obtained by Mr. P. B. Henshaw in the course of an investigation which, in so far as the chemical compositions of the materials used are concerned, covered approximately the same ground. The steels employed by the latter included a 9.58 per cent chromium steel; a 13 per cent chromium stainless steel; a 2.5 per cent silicon, 8.2 per cent chromium steel; a high-speed, 18.9 per cent tungsten steel; a cobalt-chromium steel, of a composition almost identical with that used by Messrs. Johnson and Christiansen; and a series of high nickel-chrome steels containing from 12 to 19 per cent of nickel, from 12 to 13 per cent of chromium and from 1.47 to 1.75 per cent of silicon. An account of the research in question was given by Mr. Henshaw at a recent meeting of the Royal Aeronautical Society. The experiments conducted included such determinations as the tensile strengths of the materials at high temperatures, the Brinell numbers under various conditions, the tendency of each alloy to scale, the impact values at high temperatures, and the coefficients of expansion. For carrying out the tensile tests an electric tube furnace was employed. The specimens were heated

to the required temperature and were pulled, while still in the furnace, after being allowed to soak for a period of 30 minutes. Commencing at 600 degrees C., tests were carried out at intervals of 50 degrees C. until 950 degrees C. was reached. It was not thought necessary to go beyond this temperature. As was to be expected, all the steels exhibited a somewhat rapid decline in tensile strength with increased temperature. The high nickel-chromium steels, however, registered a more gradual fall than the others, and, moreover, the drop was at a more uniform rate. While fully agreeing that when the strength of a material at high temperatures is being considered, time is an important factor, Mr. Henshaw maintains that tensile tests, obtained under the conditions described, give values comparable with those obtained in actual practice. It is well known that under prolonged heating, "creep" takes place in a material, if the stress is sufficiently high. On the other hand, a valve cools down quickly after having run hot, when the engine is shut off, and, as Mr. Henshaw points out, the properties of the valve material should be unimpaired, providing that it has not been long enough at a high temperature to have actually yielded.

All the materials tested during the course of the investigation, with the exception of the high nickel-chrome alloy steel, were found to have distinct air-hardening properties. It will be remembered that Dr. Aitchison stated definitely that a valve steel should show no liability to harden when cooled in air from the temperature which it might attain when used normally as a valve in an internal combustion engine. The Brinell hardness of the specimens after heating them to various temperatures and then allowing them to cool in air were carefully determined by Mr. Henshaw. He found that the tungsten, chromium, stainless, and cobalt chromium steels reached their lowest hardness after exposure to 800 degrees C. Air hardening sets in, however, when the specimens are cooled from 850 degrees C., and, after exposure to temperatures of from 950 and 1000 degrees C., the Brinell numbers of the steels are, in all cases except that of the tungsten steel, appreciably greater than those of the original material. On the other hand, the curve for the high nickel-chromium steel is practically a straight line until 900 degrees C. is reached, after which it falls.

The resistance to oxidation of the various steels employed was determined by exposing small test cylinders to a temperature of 1000 degrees C., in a gas furnace, under what are described as "rather adverse conditions," weighing the loss in weight due to scale, and expressing it as the loss in weight in grammes per square inch per hour. The figures given are as follows:—silicon-chromium steel, 0.0258; high nickel-chromium steel, 0.0563; cobalt-chromium steel, 0.1418; stainless steel, 0.2197; chromium steel, 0.2985; and high-speed tungsten steel, 0.3983. In the case of the high nickel-chromium steel, it is stated that the scale formed is not only slight, but is very adherent and is not likely to break away from the work. The last series of physical tests described by Mr. Henshaw dealt with the dilatation properties of his steel specimens. Experiments showed that the high nickel-chromium steel expanded and contracted almost uniformly on heating and cooling, the other steels began to diverge at a temperature of just over 500 degrees C. Furthermore, the high nickel-chromium steel has a greater coefficient of expansion than any of the other steels tested. This is interesting from the designer's point of view; the high coefficient of expansion must be taken into account in order to prevent the sticking of the valves in the guides.

Taking everything into consideration, Mr. Henshaw is of opinion that the high nickel-chromium steels possess advantages which are certainly worthy of the consideration of engineers engaged in the manufacture of valves for

high-duty internal combustion engines. The forging of this material, he says, presents no great difficulty, it depends to a large extent on governing the forging temperature rather more exactly than is the general case for other steels. The most advantageous temperature for the forging operations lies between the narrow limits of 1050 and 1100 degrees C. If the correct working temperature is adhered to, stampings may be made having a regularity of physical properties which leave little to be desired. As pointed out above, Messrs. Johnson and Christiansen stated definitely in their conclusions that steels containing from 6 to 15 per cent of chromium, together with high nickel, could not be used in the manufacture of valves for aero engines since they could not be hardened sufficiently to prevent excessive galling of the stems and wear on the tips. Mr. Henshaw furnishes an answer to these contentions, he says that the fitting on the tip of the valve of some form of hardened thimble does not offer any serious disadvantage and is regularly done with success. While admitting that all austenitic steels are liable to have a poor wearing surface, he contends that this defect can be minimized by polishing the surface of the material. He, therefore, recommends that the stems of high nickel-chromium steel valves should be given as good a surface polish as possible. He is also of opinion that the material used for valve guides should be selected with the greatest care.

The cobalt-chromium steel, recommended by Messrs. Johnson and Christiansen as being suitable for the manufacture of aero-engine valves, does not appear to have given evidence of any outstanding merit in the course of Mr. Henshaw's investigation. The latter, however, does not definitely state that this material, in common with the other steels tested, is unsuitable for valves, he merely directs all his efforts to proving that high nickel-chromium steel gives the best all-round results. He agrees that this alloy has not reached perfection, but his recommendations are strengthened by the statement that practical tests on a very large scale have proved that, under the most severe conditions, valves made of this material have worked very well indeed. His opinion that the introduction of high nickel-chromium steel for the manufacture of heavy-duty valves marks a forward step in this particular branch of metallurgy, and points out a probable line of future development, will be received with interest by all areo-engine designers.—"Engineering," December 3, 1926.

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## BEARING METALS.

The metallurgy of bearing metals forms the subject of a paper read to the Institute of Marine Engineers by Mr. W. T. Griffiths on April 20th last. This paper begins with an enumeration of the essential properties of bearing metals, followed by descriptions of the three principal groups, *viz.*, tin-base, lead-base and copper-base alloys, the effect of casting on the microstructure of these alloys, and the various tests in general use, and concludes with a short note on lubrication. Several useful tables and interesting micrographs are given.

In the first place a bearing metal must be capable of resistance to wear, and it has been found that for this purpose two constituents should be present, one in the form of hard grains which make up the actual bearing surface, and the other more plastic, to allow of uniform distribution of the load. The alloy should have sufficient compressive strength at the temperature of working to withstand the load; it should not be brittle and must be easily workable. The melting-point must not be too low, and a high thermal conductivity is desirable. These and various other properties are

mainly dependent on the chemical composition and the microstructure, which is greatly influenced by the casting temperature.

Tin, lead, copper, antimony, zinc, nickel and other metals are used in making different bearing alloys, but those in most general use fall under the three headings noted above.

*Tin-base Alloys.*—The addition of a little copper—more than 1 per cent—results in the formation of a compound harder than the surrounding matrix; the addition of less than 7 per cent of antimony gives a solid solution, while with more than 7 per cent a compound is formed. In practice, both copper and antimony are added to tin to obtain a bearing metal. A typical tough alloy made in this way has the composition tin 93 per cent, antimony 3.5 per cent, copper 3.5 per cent. If the copper or antimony content is increased, more of the crystalline compounds are formed, and the alloy becomes harder, but less tough, owing to the brittleness of the compounds. A useful alloy is that containing 86 per cent tin, 10 per cent antimony and 4 per cent copper.

As tin is expensive, attempts have been made to replace some of it by lead. This metal forms no compound or solid solution, and exists either as globules of lead or as part of the eutectic forming the matrix. The properties of the lead-base alloys are discussed below.

Zinc is sometimes added to tin-base alloys, and is said to increase resistance to rolling. The addition of nickel appears to be of no advantage, and in some cases may be actually deleterious.

*Lead-base Alloys.*—These alloys usually contain 10–15 per cent antimony, with or without tin. If no tin is present they consist of cuboid crystals of antimony in a matrix of antimony-lead eutectic. In the presence of tin, crystals of a solid solution of SbSn in tin are formed.

Lead reduces the hardness of a bearing metal, but is said to add to its anti-frictional qualities and to wear better. On the other hand, lead-base alloys are more difficult to cast owing to their higher viscosity when molten and the heaviness of lead. They are also more liable to pitting if an alkaline lubricant is used.

During the war a type of lead-base alloy was developed which contains small quantities of barium and calcium (usually less than 3 per cent), sometimes accompanied by antimony. This alloy is said to give even better results than a high-grade tin-base alloy, but has to be prepared electrolytically and is difficult to remelt.

*Copper-base Alloys.*—When a bearing metal is required which has to withstand very high pressures and heavy blows, and especially when the normal temperature of running of the bearing is high, copper is used as the basic metal. Tin, lead or zinc are added separately or together, and sometimes phosphorus.

If the rate of cooling after casting is not too slow, a copper-tin alloy containing more than 8 or 9 per cent tin consists of a solid solution, which forms the greater part, and a eutectoid mixture of this $\alpha$ solution with the $\delta$ constituent, a hard, brittle compound. The addition of phosphorus produces a sounder casting and gives rise to another hard compound of copper and phosphorus.

The copper-base alloys lack plasticity, and lead may be added to remedy this defect. The resulting "plastic bronzes," as they are termed, are cheaper and more easily worked and have a lower rate of wear.

Zinc is also sometimes added in order to produce sounder castings, but it has a bad effect on the anti-frictional and mechanical properties. It may replace tin altogether, but the resulting alloy has a low tensile strength and ductility and can only be used for light work.

The addition of nickel is of doubtful value.

*Effect of Casting on Microstructure.*—The most important factors to be considered in casting an alloy are:

(1) The highest temperature attained by the molten metal.
(2) The time during which the alloy is molten.
(3) The casting temperature.
(4) The rate of cooling after pouring.

(1) If the temperature is too high the metals oxidize; this may also lead to the inclusion of oxides in the bearing. The content of the hardening metal is lowered by oxidation, and increased wear of the shaft owing to the hard oxides, brittleness and poor adhesion to the shell result.

Hudson and Darley state that there is a critical temperature for each alloy (about 50 degrees C. above the temperature at which solidification begins) above which cooling without stirring results in an increase of the hard crystals and consequent brittleness, while below it a fine structure can be obtained.

(2) If the alloy is kept in the molten state too long oxidation again occurs, and also, owing to the tendency of the heavier metals, especially lead, to sink to the bottom, if complete solidification is delayed parts of the bearing may become of different composition and structure. Care should therefore be taken to pour as quickly as possible.

(3) The casting temperature must be high enough to allow the whole bearing to be filled while the metal is in the liquid state, but not so high as to result in the defects mentioned above. Lead-base alloys need the most careful temperature control.

(4) Too rapid cooling results in the partial or complete prevention of the separation of the hard particles, while too slow cooling results in excess of these hard particles and consequent brittleness. In the case of bronzes the rate of cooling must allow the uniform distribution of the hard $\alpha + \delta$ eutectic without the formation of a continuous network. The distribution of lead must also be carefully controlled.

Accurate temperature control by means of pyrometers is, of course, always desirable, but where only an occasional casting is made, crude methods, such as the charring of a pine chip, are used.

The shell temperature depends on the size of the shell and the amount of metal to be poured. For large bronze bearing shells 100 degrees C. is suitable. For smaller or steel shells a somewhat higher temperature should be used. The shell should be thoroughly cleaned before the metal is poured in.

*Tests.*—No really satisfactory way of testing a bearing metal has yet been found. As has been seen, various properties are all of importance, and also it is very difficult to devise a suitable test piece. Thus standardization has not become possible.

Hardness tests, preferably the Brinell ball test, are of value as giving some idea of the plasticity and tenacity and in comparing properties at ordinary and high temperatures. The hardness of the various constituents is also of importance, and to determine it a scratch hardness test has been developed by the American Society of Mechanical Engineers.

Tensile tests are not of great use, as a bearing metal is hardly ever stressed in tension.

Compression tests are very important, and standardization of these tests should be undertaken.

Impact tests are not of general utility.

Chemical analysis and microscopic examination are, as has been shown above, of the greatest value.

*Lubrication.*—One of the chief advantages of a bearing metal is the beneficial effect it may have on the preservation of the oil film in the bearing. Full knowledge of the mechanism of lubrication is urgently needed, but recent work supports the theory that the wearing away of the softer constituent results in the production of minute channels through which the oil is conveyed by something akin to capillarity.—"Supplement to The Engineer," October 29, 1926.

## THE *DOLIUS*.

If there is developing any tendency toward the standardization of the Diesel engine, it is still far below the surface and not even faintly apparent to the casual observer. Indeed evidence seems almost daily to accumulate which points quite in the opposite direction. What with small, relatively fast running engines operating through electric, hydraulic or mechanical reduction gearing, on the one hand, and on the other the increasing number of large, slow speed double acting machines, the time when the Diesel may assume in general a standard form, similar in character to that of the three-cylinder triple expansion steam engine, seems farther away than ever; certainly more distant than it did a few years ago when the four-cycle single acting engine was practically the only one in actual service at sea. While by far the larger part of ocean-going motor-ships are still propelled by the latter type of engine, the past few years have witnessed the rapid development and extensive adoption of the two-cycle engine and double acting engines of both two and four-cycle types are already in use at sea and seem likely to become the rule in large vessels of relatively high power. On the other hand the small non-reversible engine, operating through some form of reduction gear, has established a firm foothold on small vessels and its promotors are not lacking in confidence that as time goes on its use will become general on large sea-going craft of various types. Altogether, surface indications at least point to an increasing variety rather than to standardization in the types of internal combustion engines employed in ship propulsion.

Perhaps the farthest excursion afield from what may possibly be considered the orthodox type of Diesel engine, so far as such a type has, up to the present time, been developed, is to be found on the most interesting vessel, from an engineering point of view, which has recently been seen in the port of New York, the motorship *Dolius*. As most readers will doubtless remember, this vessel, or particularly her propelling machinery, was given much attention by this and other technical papers when she first appeared in service, something over two years ago. The recent visit of the *Dolius* to New York, on her way to the Far East, aroused renewed interest in local circles and many engineers availed themselves of the opportunity of visiting and inspecting what is undoubtedly one of the most unique and interesting ships at present afloat.

The *Dolius* is a general cargo vessel about 400 feet in length and 8000 tons deadweight capacity, propelled by twin screws. The propelling machinery, upon which interest in the vessel mainly centers, consists of two Scott-Still combined steam and internal combustion engines of about 1250 shaft horsepower each. These engines are each provided with four cylinders, the upper parts of which operate on the Diesel two-stroke cycle and the lower as single acting steam engines. The steam is generated in the cooling jackets of the Diesel cylinders and in two exhaust gas, watertube boilers arranged in series with the cylinder jackets. A fact at first somewhat difficult to grasp is that the pistons are *cooled* by the steam acting

on their lower sides. Steam first enters the after cylinders at a pressure of 140 pounds per square inch. Coming in contact with the relatively very hot piston it is superheated and then used expansively in the remaining three forward cylinders. Exhausted from these, but still retaining the ability to perform useful work, it passes through a low pressure turbine which drives the blower provided for scavenging the two-cycle Diesel cylinders and finally finds its way to the condensers where a vacuum of 28 inches is maintained. Going to the other end of the chain it is found that the exhaust gases not only pass through the steam boilers already mentioned but after leaving the latter are passed through a feed water heater before final exhaust to the atmosphere, by way of the funnel. So far as we are aware no other heat engine has ever been devised, certainly none has been put to practical use, which converts to useful work such a large proportion of the heat generated by the fuel as those of the type installed in the *Dolius*. At the time of her visit to New York the vessel had traveled slightly over 120,000 miles and her average fuel consumption figured out 0.367 pound per shaft horsepower, a fuel economy which, it is claimed, has never been equaled in sustained service.

The practical engineer, while perhaps ready to admit the superior economy, will be at least equally concerned with the questions of operation, upkeep and repairs. For him the statement of the chief engineer that repairs had been practically nil and that the auxiliary Diesels of ordinary type required far more care than the main engines, is of particular interest. It must be owned, however, that a superficial inspection of the *Dolius*' machinery gives the impression of considerable complication. While no cams and valve gear are required for the Diesel portion of the engine, the valves for the steam ends are operated hydraulically by oil which sounds somewhat formidable, but is reported to function perfectly and to give no trouble whatever. Maneuvering is performed by steam and as solid injection of fuel is used no air compressors are needed, in itself a simplification of importance. Furthermore, the compressed air used to maneuver an ordinary Diesel ceases to function once the engine is fairly started and, so far as service operation is concerned, is a direct loss, while in the Scott-Still engine the steam not only starts the engine but continues at all times to increase its power and very materially adds to its economy of operation.

As in all the problems of commercial engineering, the solution in the present case depends in the last analysis upon the sum of initial and operating costs and its comparison with that of other prime movers. That the Scott-Still engine is economical in the use of fuel there can be no doubt and so far as can be judged from the performance of the *Dolius* the cost of upkeep and repairs is not in excess of that of other types of machinery. As to first cost, we have no knowledge but it must be owned that upon the surface at least the installation on the *Dolius* looks rather complicated and expensive. Only by long periods of commercial service in more than one vessel can really definite conclusions be arrived at, but in the meantime the interesting and significant fact should be noted that the owners of the *Dolius* now have under construction one or more new vessels in which Scott-Still engines are to be installed.—" Marine Engineer and Shipping Age," December, 1926.

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## THE DEVELOPMENT OF THE MINE-LAYING CRUISER.

Seldom has a ship been built behind a greater veil of secrecy than the new mine-laying cruiser H.M.S. *Adventure*, which was launched at Devonport in June, 1924, and is now undergoing her trials. She has a displacement of

7260 tons on an overall length of 520 feet, a beam of 58 feet over her bulges, and a mean draught of 19¼ feet, and there is reason to believe that she has a speed of 27.75 knots and four 4.7-inch guns. Beyond that nothing is known of her, and the wisdom of the Admiralty's policy of secrecy is shown by the fact that the French have held up the construction of their surface mine-layer *Pluton* "on account of the rapid development of this type"—in other words, until they see what features the design the *Adventure actually embodies.*

The action of the British Admiralty in building a ship costing over a million pounds for the purpose of laying offensive minefields marks a very radical departure from the policy of fifty years. Until Germany showed during the war that mine-laying could be very effective as a form of offense, it was rather looked down upon in the British Navy, and it was some time before the Service was persuaded to take any part in it at all. For many years it was put into the hands of the Royal Engineers, who maintained a fleet of small steamers which were designed for mine-laying, but were only suitable for inshore work, at entrance of the ports to be defended. The mine, in fact, was regarded as essentially a defensive weapon, a weapon only suitable for a Navy whose strategy was passive by nature. As an object-lesson, the enthusiasm of the Russian Navy for mine-laying, and its well-known preference for staying in port, was pointed out by every strategist.

It is true that Russia was the first country whose navy took to the mine really seriously and brought it to any pitch of perfection. It is probable that in mining work they led Europe right up to the time of the 1917 revolution, and it is possible that some of the mines that were laid during the Czarist régime have been swept for ineffectively many times and are still capable of doing damage. It was, accordingly, only natural that the first European men-of-war to be designed especially for mine-laying should fly the Russian flag—the *Yenesei* and *Amur*, which were built at the end of the last century. They were not merely mine-layers, as they were also designed to act as torpedo transports for the supply of the fleet. They were useful vessels with a displacement of 2500 tons and a nominal speed of 17½ knots, although actually neither of them got within a knot of it in service. They were armed with five 12-pounder quick-firers, and a number of smaller pieces which in those days were regarded as being quite sufficient to beat off any destroyer attack. The *Yenesei* was blown up by one of her own mines off Port Arthur at the beginning of the Russo-Japanese War, and this put the whole type under something of a cloud. The disaster was, however, entirely due to the carelessness of the commander in taking cross-bearings.

Germany was the only country which refused to be influenced by this incident, and they carried out a number of experiments with their transport *Pelikan*, which was specially reconstructed for mine-laying. She had a displacement of 2300 tons, at her best a speed of 15 knots, while she was armed with four 15-pounder quick-firers. As soon as she maneuvered with the fleet it was realized that her speed was deficient, and accordingly she was used for experimental purposes only, and was largely responsible for the quality of the German mines in the early days of the war. After the experience gained with her, the German Admiralty built the *Nautilus* of 1906—a ship of just under 2000 tons with a speed of 20 knots and an armament of eight 22-pounder guns. Her deficiency in speed and gunfire was shown by the fact that she was driven ashore on the Swedish coast by the Russian fleet and afterwards had to be interned. She was followed a year later by the *Albatross*, a ship of a slightly larger displacement but the same speed and gun-power. In her internal design, however, she was a great im-

provement of her predecessor, and carried 600 mines against the 200 of the earlier ship.

The experience of the war soon showed that these German mine-layers were too slow and ill-armed for their work, and that the difficulty of convoying them on any offensive operation would be too great for their employment. Accordingly, for laying mines on the British coast, the cruiser *Kolberg* was used, a ship with a speed of 25½ knots, and an armament of twelve 4.1-inch guns. She and many other cruisers in the German, Austrian and Italian Navies had been fitted for the carriage of a large number of specially-designed mines. As far as the actual operations were concerned she was satisfactory, but her use for this purpose was most unpopular in the Navy, where the officers maintained that a cruiser should be a cruiser, and that they had none too many ships of the type for the duties that they would be called upon to perform. As for carrying out cruiser work while she had her mines on board, the risk of a colossal explosion was obviously too great. The French also soon realized, in the test of war, that the two mine-layers that they had built—the 20-knot *Cerbere* and *Pluton*—were of little use except for defensive work, which might easily be done by tugs or trawlers. These ships had a displacement of only 600 tons, but they carried 140 mines of a special design.

When the war broke out, and it became obvious that mine-laying would have to be carried out on a big scale, it was to auxiliaries that the combatant navies turned. The *Konigen Luise*, a 20-knot channel steamer, succeeded in drawing the first blood in the mining war, although she was soon destroyed when she fell in with a British flotilla. However, she had proved that she could do certain work in certain circumstances, and the Germans commissioned a number of channel steamers as mine-layers. They also made use of the train ferries which usually maintained the service to Sweden. They were deficient in speed and armament, and were only suitable for laying defensive fields, but the railway lines which were already laid on their main deck and the open sterns made it possible for them to lay fields far more quickly than any other ship, given the necessary naval protection. The British also used channel steamers, and a certain number of small liners. Generally speaking their speed was satisfactory, but their fuel capacity was poor, and they had the disadvantages which always attend the conversion of a merchantman into a man-of-war. Had the command of the sea not been so thorough, the casualties in this branch would probably have been very considerable.

It was Germany which first considered the idea of using submarines for offensive minefields, and those that they laid in the Downs, and on the trade routes, caused a good deal of confusion to British shipping. Their essential disadvantage was that they suffered from a small radius of action, and it was in attempting to remedy this that the German Admiralty built a class of submarine which shot out its mines after the fashion of a torpedo. These submarines were always giving trouble, but in the destruction of H.M.S. *Hampshire*, with Lord Kitchener on board, by *U 75*, a boat of this type had an important success. Towards the end of the war Germany evolved a number of types of submarine mine-layers, but, as their mine-laying equipment necessitated the curtailment of the torpedo equipment, they were only of use against merchantmen. For mining the Heligoland Bight, the British Navy used a number of submarines of the *E type*, but this involved the sacrifice of the greater part of their fuel by fitting mine tubes in place of several of their saddle tanks. Like most submarine mine-layers up to the present time, they proved that they could be exceedingly useful at short-distance work, but they could not hope to cover a large area.

Practically all the navies of the world have small mine-layers that are capable of doing useful work in coastal waters, and several of them have followed the British lead in converting a number of cruisers that were too old for their real work, but which can cover a large radius of action. The scientific manner in which the Germans laid their mines showed that offensive operations could only be carried out by a ship that was practically perfect for her purpose, and such a ship cannot be obtained unless she is specially designed. Even on the meager facts that the Admiralty has published, the cruiser mine-layer *Adventure* is obviously an important example of her type, and when the authorities choose to divulge her details they will be followed with the very keenest interest in every navy. In the meantime it is understood that she is to undergo a long series of trials, and that a large number of problems, which have hitherto been worked out only in theory, will be settled as definitely as they can be in peace time.—"Engineering," December 24, 1926.

## CONDENSER STUDY SHOWS BAD WATER DISTRIBUTION.

By John J. Grob and N. Artsayooloff,
United Electric Light & Power Company.

Over a year ago temperature exploration tests were conducted at Hell Gates Station on a 50,000-square-foot surface condenser with radial steam flow, two water passes and divided sections, each half supplied by a separate circulating pump with a capacity of 35,000 gallons per minute. Owing to certain interesting indications, it was decided to explore these conditions somewhat further, and a series of temperature and velocity tests were planned and carried out.

In order to develop a comprehensive picture of the changing conditions on the water and steam sides of a large condenser, it would be necessary to investigate distribution of water among the tubes, distribution of condensation work among the tubes, and steam pressure and temperature drops across the tube bank. The first two investigations have been completed on the condenser and form the basis of this article.

Figure 1 is a tube-sheet layout for the condenser selected for test. A number of directions radially across the tube sheet in the reversing chamber were selected for investigation and were labeled directions A, B, C, D and E.

It was necessary to construct stuffing boxes and connect them at the end of a nipple and valve on the chosen directions so that a tube could be inserted through the stuffing box and moved across the tube sheet without danger of losing the condenser siphon.

Temperature explorations were made with resistance thermometer elements encased in $\frac{5}{8}$-inch metal tubes about eight feet long and connected to an instrument capable of indicating differences of temperature up to twenty degrees. One bulb was inserted in the inlet chamber, a second one in the tail pipe and a third one was traversed over a series of points along the various directions selected for investigation. Hence the temperature differences were measured with the inlet temperature of the cooling water as a base. Although the load on the unit was maintained fairly steady during the tests, the small fluctuations that did occur affected the total rise, therefore the temperature rise in every investigated point was referred to the simultaneous total rise in cooling-water temperature. Additional connections in the outlet chamber of the second pass disclosed the variations in the total rise across the second pass.

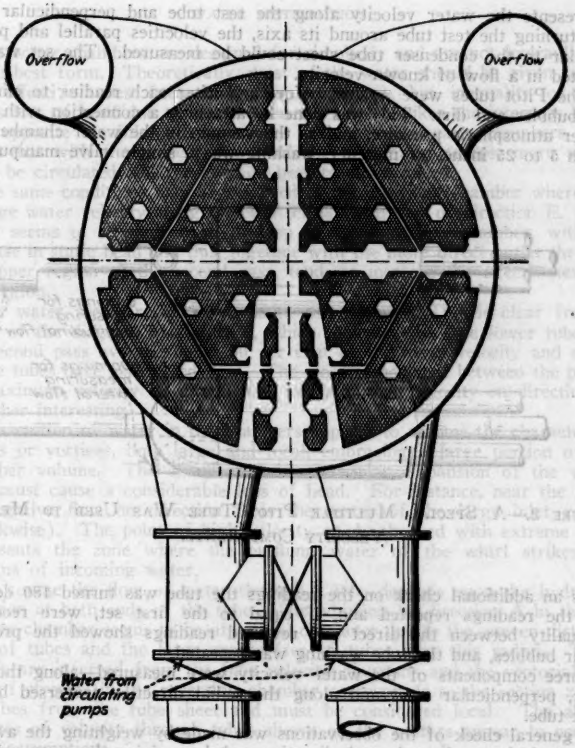


FIGURE 1.—THE INLET AND OVERFLOW WATER BOX IS DIVIDED INTO QUARTERS ON THE VERTICAL AND HORIZONTAL DIAMETERS.

Right and left halves are independent, with separate pumps, to facilitate cleaning. The division valve is usually kept closed. In each half, the lower quarter is the first pass.

The apparatus proved to be very sensitive, responding in 10 to 20 seconds to the small changes in temperature, but the readings represent the mean of approximately eight tubes just around the point investigated.

For the purpose of measuring velocities at the various points to determine the distribution of cooling water among the tubes, a special Pitot tube was made up consisting of four copper tubes ⅛ inch inside diameter soldered in a ⅝-inch brass tube about 8 feet long. These tubes were arranged as shown in Figure 2. Thus an instrument was obtained that could transmit a velocity head to a U-tube for measurement. From the sketch of the improvised Pitot tube it is clear that a velocity head can be measured which

represents the water velocity along the test tube and perpendicular to it. By turning the test tube around its axis, the velocities parallel and perpendicular to the condenser tube sheet could be measured. The set was calibrated in a flow of known velocity.

The Pitot tubes were washed before and after each reading to eliminate air bubbles and dirt. This was done by attaching a connection with water under atmospheric pressure, and as the vacuum in the water chambers ran from 5 to 25 inches of mercury, washing was a simple valve manipulation.

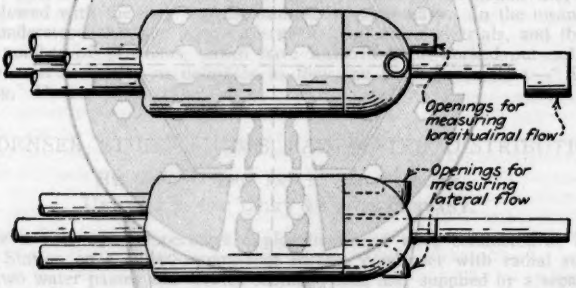


FIGURE 2.—A SPECIAL MULTIPLE PITOT TUBE WAS USED TO MEASURE VELOCITY COMPONENTS.

As an additional check on the readings the tube was turned 180 degrees and the readings repeated and, if equal to the first set, were recorded. Inequality between the direct and reversed readings showed the presence of air bubbles, and the tube washing was repeated.

Three components of the water velocity were measured—along the tube sheet, perpendicular to it and along the radial direction traversed by the Pitot tube.

A general check of the observations was made by weighting the average of the velocity readings in each direction and determining the mean velocity. The mean velocity determined in this way was found to be 7 feet per second. The velocity computed by dividing the amount of water by the total tube passage area is 7.07 feet per second. This appears to indicate a fair average degree of accuracy.

The average distance of the Pitot tube orifices from the tube sheet was 2½ to 3 inches, affected somewhat by bending of the test tube. It was not feasible to make the observations any closer to the tube sheet. Upon moving the test tube slowly, one could not detect any sudden change in velocity head, hence the individual jets of water from the tubes were already joined in a uniform flow at this distance from the tube sheet.

Unfortunately, the interference of piping, columns and girders prevented the use of a longer Pitot set and also made it impossible to investigate direction $E_4$ in the second pass. and as a basis for the determination of an average velocity and condensation work on this direction, the total flow determined in the first pass was used in conjunction with the weighted averages in the several measured directions in the second pass.

## WATER DISTRIBUTION.

The water chambers are designed for simplicity and are not hydraulically of the best form. Theoretically, most of the water should be given to the most effective tubes (near the steam entrance), but inequality of flow causes loss of head due to shock when the water strikes the interpass bulkhead opposite the inlet. By properly designing the water chambers it is likely that with the present power of the circulating pumps, more water could be circulated and the average heat transfer raised.

The same conditions may be observed in the reversing chamber where the average water velocity along direction $E_1$ is twice that of direction E. The water seems to rebound from the top of the reversing chamber, with an increase in static head and this, together with the more direct outlet through the upper region of the second pass, tends to increase the effectiveness of this region.

The water seems reluctant to make sharp turns. This is clear from a study of the diagram of Figure 3, which indicates that the lower tubes of the second pass average about 50 per cent of the mean velocity and many of the tubes have a reversed flow. The close coincidence between the points of maximum velocity on direction C and minimum velocity on direction D is rather interesting.

The motion of water in the chambers appears to assume the character of whirls or vortices, both large and local, embracing a large portion of the chamber volume. The chambers being irregular, expansion of the water flow must cause a considerable loss of head. For instance, near the point of negative flow on direction B is the center of the large inlet vortex (clockwise). The point of high velocity on A, checked with extreme care, represents the zone where the outflung water of the whirl strikes the streams of incoming water.

Of course the flow of water through a tube depends upon the hydraulic situation at both ends of the tube. The whirling on direction A in the reversing chamber seems to be of a local character due to the uneven distribution of tubes and the large spaces free of tubes near this radius. These whirls are inferred from the tube velocity diagrams and the vectorial picture of streams in the reversing chamber corresponds to the section 2½ to 3 inches from the tube sheet and must be considered local. The abrupt changes of velocity through tubes shown on the vectors appear to confirm these assumptions.

Practically everywhere the water velocity through the outermost tubes located on the circumference is relatively low. This may be accounted for by the following considerations: Along direction A, the inlet ends of the tubes are in the place of greatest tangential velocity of inlet water and the head available to overcome the resistance of the tubes is reduced by the velocity head of the inlet water and so the velocity through the tubes is low. Along directions B and C there is back pressure in the reversing chamber in the turbulent zone agitated by the frictional resistance of the chamber walls and the bare spaces on the tube sheet. The magnitude of the back pressure is approximately the average velocity head in the reversing chamber. In the zone of friction the water is relatively almost at rest. On direction C this back pressure is overcome by the conversion of velocity head in the inlet chamber into static head when the water rebounds from the top corner of the first pass. Along direction D the water is reluctant to make sharp turns, and as the entire body of circulating water flows past this point, the velocity is high. Near direction E the tail pipe of the condenser is attached at the side of the upper part of the outlet chamber. Therefore the

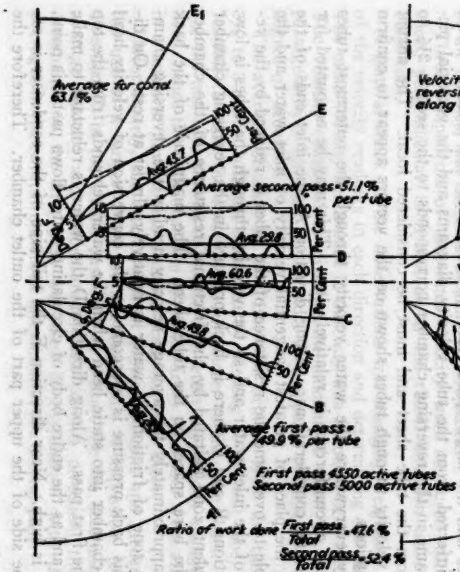


FIGURE 3.—TEMPERATURE RISE AND HEAT ABSORBED PER TUBE.

Dotted curves show local temperature rise referred to 10 deg. average rise for the entire condenser, and full curves show heat absorbed per tube in per cent of the average for all tubes.

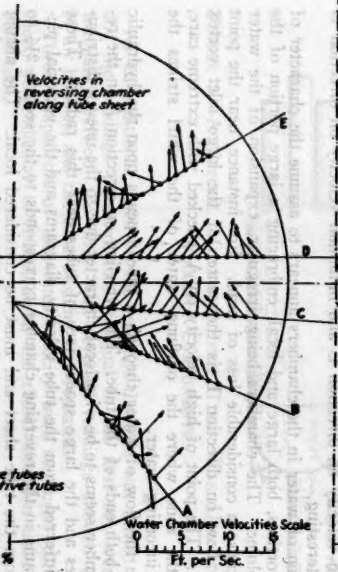


FIGURE 4.—WATER VELOCITY PARALLEL TO THE TUBE SHEET.

The arrows show the direction of flow, and by their length the velocity, to the scale at the bottom. It is evident that flow is very irregular.

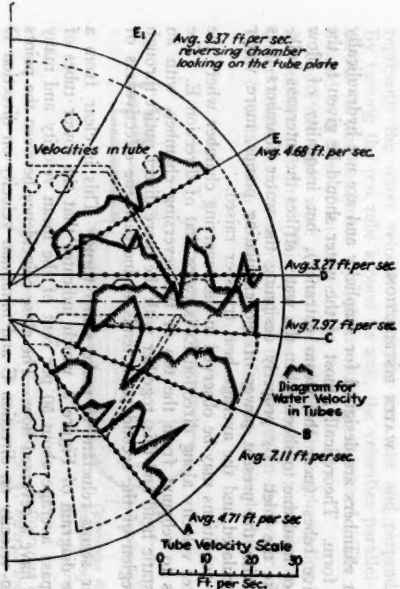


FIGURE 5.—WATER VELOCITY IN GROUPS OF TUBES.

The shaded curves indicate velocity by their perpendicular height above the radii on which they are plotted, for groups of eight tubes centering around the dots on the respective base radii.

greatest outlet velocities into the outlet chamber should be in the central partition and the upper corner. On direction E velocities would tend to be low near the circumference.

The water velocity diagram gives a complicated flow picture, and the analysis of it is difficult because both inlet and outlet sides of the passes must be considered simultaneously. The four deductions outlined above lead to a congruent picture. It is confirmed again in the exceptional high tube velocity on direction $E_1$ which was not observed by test, but was calculated to make up the average for the entire second pass. Increase of head in the top corner of the reversing chamber due to rebound from the top and a decrease in the outlet chamber due to the immediate suction of the condenser tail pipe makes this part of the condenser most efficient and the calculated drop appears to be correct.

With a siphon circuit under high vacuum there is always a certain amount of air leakage into the system.

Some dissolved air will also be liberated from the circulating water. The presence of such air was detected in the tail pipe of the condenser. The tail pipe is about 3 feet 6 inches in diameter, and in the center was found a stream about 6 inches in diameter where the temperature varied abruptly and irregularly over a range of about 10 degrees. A Pitot tube inserted into this region vibrated as if under the impact of blows. Inserting the tube farther into the tail pipe diminished the vibration until on full insertion it ceased altogether. The presence of air in the circulating water increases the velocity of the water through the tubes and the resistance of the passage. On the other hand, it increases the turbulence of the water, and on the whole the net effect of air in cooling water, so far as heat transfer is concerned, is probably insignficant.

### NON-UNIFORM DISTRIBUTION OF CONDENSATION WORK AMONG TUBES.

The first impression is that the diagram in Figure 3 appears to show a close relationship between condensation work and tube velocity. This result should perhaps be expected. The average condensation per tube of the first pass is 49.9 per cent and of the second pass 50.1 per cent. The water velocity through the second pass is, however, lower and so too is the mean temperature difference. There is therefore some inherent difference in heat transfer between the first and second passes. The effect of air may be omitted in this case because steam is admitted on the whole circumference. The so-called air cooler must of course be considered integrally with the condenser. Therefore the major cause of difference must be the insulating effect of falling condensate, which adheres to the tube surfaces and blankets them from the steam.

### NUMERICAL EVIDENCE FROM THE TEST DATA.

Conditions were approximately as follows: Temperature of water: Inlet, 60 degrees; outlet, 70 degrees; inlet to second pass, 65 degrees; steam temperature, 85 degrees; arithmetical mean temperature difference first pass, 22.5 degrees; second pass, 17.5 degrees; ratio of water velocities, first to second pass, $5000/4500 = 1.11$. Therefore the ratio of heat transfer averages in second and first pass will be

$$\frac{22.5}{17.5} \times \sqrt{1.11} = 1.35$$

This means that the loss of heat transfer from the value for the top tubes of the second pass to the value for the bottom tubes in the first pass is at least 59 per cent. This loss is apparently due to the effect of dripping condensate.

A further effect of the insulating action of condensate is manifested in the drop of condensation work for tubes at the circumference. Immediately

TABLE 1.—COMPARISON OF AVERAGE HEAT TRANSFER FOR ALL TUBES WITH FIGURES FOR PERIPHERAL TUBES AND FOR TUBES JUST INSIDE THE PERIPHERY.

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| First Pass | Average on $B$ | $\frac{49.8\%}{\sqrt{7.11}} \times \frac{17.5}{22.5}$ | = | 14.6 | | | |
| | Average on $C$ | $\frac{60.6}{\sqrt{7.97}} \times \frac{17.5}{22.5}$ | = | 16.7 | Average | 18.0 | |
| Second Pass | Average on $E$ | $\frac{43.7}{\sqrt{4.68}}$ | = | 20.2 | | | |
| | Average on $E_1$ | $\frac{63.1}{\sqrt{9.37}}$ | = | 20.6 | | | |

| The Peripheral Tubes | | | | | Most Efficient Tubes (Next to Peripheral Ones) | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $B$ | $\frac{25}{\sqrt{3.0}} \times \frac{17.5}{22.5}$ | = | 11.2 | | $B$ | $\frac{83}{\sqrt{9.5}} \times \frac{17.5}{22.5}$ | = | 21.0 | |
| $C$ | $\frac{65}{\sqrt{6.8}} \times \frac{17.5}{22.5}$ | = | 19.4 | | $C$ | $\frac{103}{\sqrt{10.9}} \times \frac{17.5}{22.5}$ | = | 24.3 | |
| $E$ | $\frac{16.0}{\sqrt{2.7}}$ | = | 10.3 | | $E$ | $\frac{69}{\sqrt{9.1}}$ | = | 22.8 | |

after these inefficient peripheral tubes are the most efficient tubes. The higher water velocity must, of course, be taken into consideration, therefore the ratio of condensation work to the square root of water velocity will be the criterion. For the first pass it will be multiplied by the ratio of the mean temperature differences to make the figures comparable.

The quality of the exhaust steam may be estimated to be about 85 per cent. The moisture of the exhaust steam is deposited on the peripheral tubes which act as a separator for steam at high velocity. The saturated steam, after passing these tubes, probably dried a bit more by some throttling on entering the tube bank, is easily condensed until it loses velocity and temperature with developed partial air pressure. It is a point of interest that the peripheral tubes on direction C which are shielded by a local baffle from the falling condensate, do not show so great a reduction of heat transfer. These tubes are presumably dryer than the average.

Theoretically, almost an equal quantity of steam must enter every square foot of entrance area of the tube bank. Actually, there are variations in the steam flow, if indeed it is radial flow, around the circumference. The steam, therefore, has preferential lanes in entering the bank of tubes. The center toward which the steam is flowing lies slightly above the axis of the condenser, at approximately the level of the staybolts. This condition has a detrimental effect on condenser performance, increasing the resistance to steam passage and reducing the exhaust vacuum.

A point of peculiar interest showing apparent freaks in steam flow may be observed at the innermost tubes investigated on direction E. At the fourth point from the end the condensation is extremely good ($63/25.8\sqrt{5.8}$

= 26.1) despite the relatively low water velocity. This can be explained by the relatively fresh steam entering around the baffles which close the vertical passage between the north and south tube banks of the condenser.

### CONCLUSIONS.

An investigation of this character, in addition to disclosing facts of purely engineering interest of possible value for future designs, is useful also to the plant engineer interested in apparatus improvement. Many of the condensers in operation today were designed to contain the maximum cooling surface in a shell of smallest possible dimensions. The steam passages are choked or restricted excessively and such a condition could possibly be remedied during a retubing period by the omission of tubes.

A study of this nature may show that by the removal of some 10 to 15 per cent of the tubes, the steam passages will be enlarged, with a consequent reduction in resistance, and by simple calculations it may be shown that, for example, a reduction of tubing on the circumference of a radial-flow condenser will improve the apparent heat transfer and the vacuum. By apparent heat transfer we mean the heat transfer calculated with the steam temperature taken at the exhaust. The actual steam temperature through the condenser is lower. A loss of 5 or 6 per cent of cooling surface represented by 10 or 15 per cent of the less active tubes will be repaid with interest in the increased steam temperature and consequent increased mean temperature difference. Submitted to such tests, many condensers might also show possibilities for improvement on the water side. The addition of one or two baffles properly located in the water chambers might distribute the water more evenly and put into active work tubes having poor circulation of cooling water.—"Power," November 9, 1926.

## APPLICATIONS OF RADIO IN AIR NAVIGATION.

By J. H. Dellinger,* Washington, D. C.

Radio is being used for many purposes, some of them more appropriate than others. The major uses for which it is appropriate include broadcasting, transoceanic telegraphy and telephony, communication with ships at sea, and communication with aircraft. The applications to aircraft are now going on with great rapidity and are similar in principle to the marine uses but very different in the actual working out. Both for marine vessels and for aircraft, radio provides the ideal means of exchanging messages, of providing navigational aids, and of remote control. There is practically no field of usefulness for remote control except in military practice, and it will be left out of consideration in this discussion.

In its important functions of carrying messages and of providing aids to navigation, there are clear lines of differentiation in the evolution of radio for marine and for air use. These depend largely on the essential differences in speed, size, and weight limitation. Another difference is the special problem of landing which the airplane offers. Whereas an airplane is a small, light, rapidly moving object, a marine vessel, from the radio viewpoint, is but little different from an island. It can carry any type and amount of radio apparatus, and can be as leisurely as it pleases in using it. Lighter-

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than-aircraft are intermediate between the airplane and marine vessel in their radio requirements. They can use any of the radio aids that an airplane can, and some of those adapted to marine vessels in addition. Airplanes are principally in view in this discussion, particularly because the problems which require attention at the present time arise out of the need of developing radio aids to airplane navigation on the existing and rapidly developing civil airways.

It is generally agreed that the primary and most urgently needed application of radio in air navigation is simply that of communication. The great problem of the airplane at the present time is that of landing. The maintenance of communication so that the pilot, regardless of weather conditions, may be kept advised of suitable landing places, is the first requisite. Comparing airplane communication with that applicable to marine vessels, it may be stated that aircraft require telephony instead of telegraphy, and that the communication be at higher frequencies, and, generally speaking, over shorter distances than for marine vessels. The use of continuous-wave radiotelegraphy is inherently a more economical method of communication than radiotelephony, and is the natural means of communication for ships. This, however, requires the use of an operator trained in the telegraphic code. Most airplanes now have no crew other than the pilot, and this condition will doubtless be true for a long time to come. It is not to be expected that the pilot will also be a telegraph operator, and it therefore follows that radiotelephony rather than radiotelegraphy will be the system used on aircraft. Higher frequencies will be used than in marine communication because of the relatively small size of airplanes. It is inconvenient and undesirable to use long antennas on airplanes. Short antennas work most efficiently with high frequencies. It is therefore to be expected that relatively high frequencies will become standard for aircraft radiotelephony.

It was stated above that aircraft communication is characterized by shorter-distance work than marine communication. This is true both because of the weight limitation on transmitting or receiving apparatus carried on the airplane, and also because of the interference to reception produced by the airplane-engine ignition system. A shielding system can and should be used on all airplane engines which largely eliminates this interference, but the perfection of this shielding is limited by the additions to the total weight. It may be said, furthermore, that there is no good reason why great distances need to be covered in aircraft communication. Here we have an essential advantage over marine navigation. On the airways of the country radiotelephone transmitting and receiving stations can be placed at convenient and relatively short intervals. These will be advantageous for the aircraft, and incidentally such a system will be more economical than one of a very few stations of great power placed great distances apart. In this respect aircraft communication has an advantage over marine communication.

It is generally agreed that communication to the aircraft is much more important than communication in the other direction. It will add immeasurably to the peace of mind of the pilots and the safety of flying when all airplanes carry radio receiving sets and there is an adequate system of ground stations telephoning information as to weather, landing conditions, etc. As the airways develop and passenger carrying becomes more common, there will be an increasing use of radiotelephony from the aircraft as well. Two-way communication will of course be a great advance. There are a great variety of technical problems involved in this and all the other applications of radio to air navigation, some of them requiring research and engineering work yet to be undertaken. The author can do no more than

mention a few of those concerned in the proper development of telephony to and from aircraft; these include the technique of shielding ignition systems, the choice of type of receiving sets, the type of power supply for transmitting sets, and the all-important problem of the assignment of frequencies.

Besides solving the problem of communication with aircraft, it is obvious that radio furnishes an instrument of navigation. During clear weather

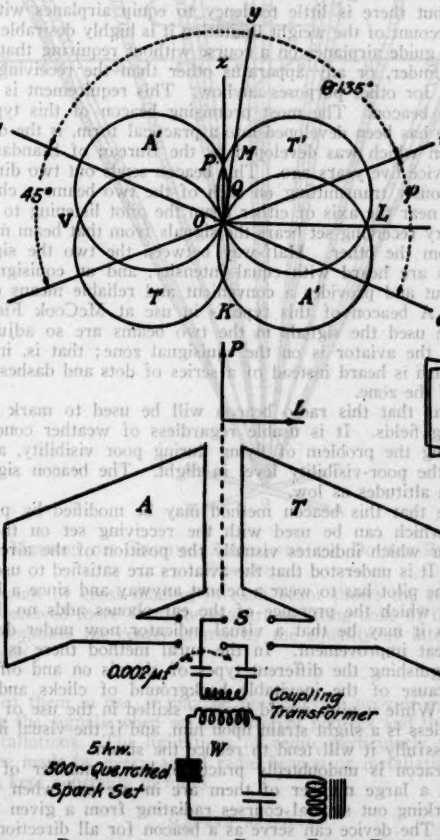


FIGURE 1.—PRINCIPLE OF THE DOUBLE-BEAM DIRECTIVE BEACON.

(In the upper part of diagram the A and T pairs of circles show in polar coördinates the relative intensities of radiation for each transmitting coil A and T in all directions. On the bisector O X the intensities from the two coils are equal.)

and for flying below the clouds a pilot needs no navigational instrument other than his eyes, for he can see the ground in the daytime and the airways will be provided with lights for night flying. When the ground is obscured by fog, however, or when the aviator wishes to fly at great elevations, radio is the only navigational aid available. In making use of the directional properties of radio there is a marked difference between the trend for air navigation and that for marine navigation. Marine vessels are finding the radio direction finder of the greatest assistance as an aid to navigation, particularly in fog, but there is little tendency to equip airplanes with direction finders. On account of the weight limitation it is highly desirable that means be provided to guide airplanes on a course without requiring that they carry any direction finder, or any apparatus other than the receiving set which will be carried for other purposes anyhow. This requirement is met by the directive radio beacon. The most promising beacon of this type, and the only one which has been developed into a practical form, is the double-beam directive beacon which was developed by the Bureau of Standards for the Army Air Service five years ago. This beacon sends out two directed radio beams, continuously transmitting on each of the two beams a characteristic signal. When near the axis of either beam the pilot listening to the signals with an ordinary receiving set hears the signals from that beam much louder than those from the other. Half-way between the two the signals from the two beams are heard with equal intensity, and an equisignal zone is thus marked out and provides a convenient and reliable means of flying a given course. A beacon of this type is in use at McCook Field, and in the form there used the signals in the two beams are so adjusted as to interlock when the aviator is on the equisignal zone; that is, in this zone a continuous dash is heard instead of a series of dots and dashes which are heard when off the zone.

It seems clear that this radio beacon will be used to mark the routes between landing fields. It is usable regardless of weather conditions. It assists in solving the problem of flying during poor visibility, as the flier can go above the poor-visibility level in flight. The beacon signals serve as well at high altitudes as low.

It is possible that this beacon method may be modified by providing a simple device which can be used with the receiving set on the aircraft, having a pointer which indicates visually the position of the aircraft on or off the course. It is understod that the aviators are satisfied to use the aural method, since the pilot has to wear a helmet anyway and since a helmet has been devised in which the presence of the ear phones adds no discomfort. In spite of this it may be that a visual indicator now under development would be a great improvement. In the aural method there is some difficulty in distinguishing the different types of signals on and off the equisignal zone because of the inevitable background of clicks and noise of various kinds. While a pilot would become skilled in the use of these signals, it nevertheless is a slight strain upon him, and if the visual indicator is developed successfully it will tend to reduce the strain.

This radio beacon is undoubtedly practical, but a number of problems will arise when a large number of them are in use and when there is a necessity of marking out several courses radiating from a given airport or beacon station. The device can serve as a beacon for all directions by continuously rotating the equisignal zone and employing a stop watch on the aircraft, but this would be a complicated system and would be so much less simple in use that its utility is doubted. By rotating the equisignal zone in steps, however, setting it for brief periods successively on the several courses desired, a satisfactory service can probably be worked out. When there

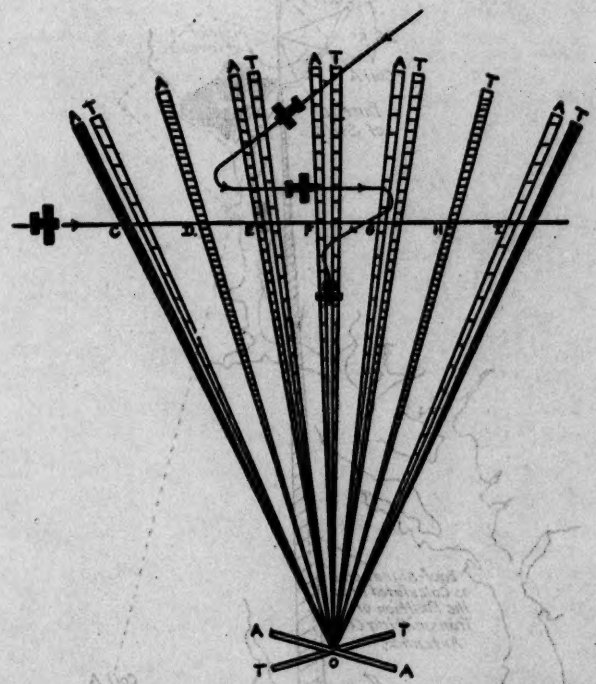


FIGURE 2—METHOD OF GUIDING AIRPLANES BY DOUBLE-BEAM DIRECTIVE BEACON.

(The amount of cross-shading in each beam shows the relative intensity of signals received there from the two transmitting coils of the beacon. The airplane recognizes the center course by the equality of intensity of the A and T signals.)

are many of them in use, it may be necessary to provide special means of recognizing the signals when set on different courses and those of different beacon installations. The limitations of radio, particularly the additions to interference made by every added radio beacon or installation of any kind, need to be kept constanly in mind.

### MARKER BEACONS.

A directive beacon of the type described is sufficiently well developed to insure that it will be used on the civil airways of the United States. It serves effectively to guide the aviator along his course. It is subject to the limitation, however, that it gives no indication as to the distance covered or as to the particular part of the route which is passed at any given time.

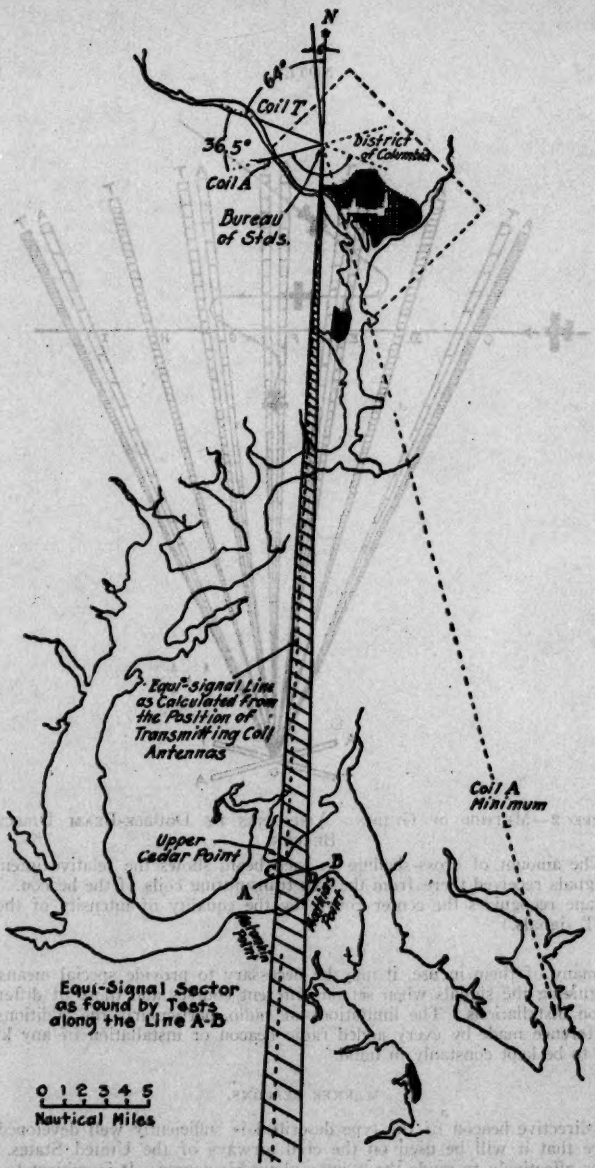


FIGURE 3.—RESULTS OF ORIGINAL TRIAL OF DIRECTIVE BEACON.

[A ship in the Potomac River located the beam as shown, and found the zone of equal signals to have the width CD. Coils A and T consist of two single-turn rectangular coils 12.2 × 35.7 meters (40 × 150 feet) crossed at an angle of 143.5 degrees. Type of transmitter, 2-kilowatt 500-cycle quenched-spark; coil current, 9 amperes; wave frequency, 300 kc. (1000 m.)]

This lack can be remedied by the establishment of very low-power radio transmitting stations at intermediate points to serve as localizers or markers of those points. These low-power non-directional beacons will give a characteristic signal which will be heard and recognized by the aviator during a short interval when he is near and over these beacons. They will not mark a line of direction but simply mark a distance somewhat as mile posts on an ordinary road. Very simple radio transmitting outfits are being designed for this service, and it is believed that they will supplement the directive radio beacon in a most effective way.

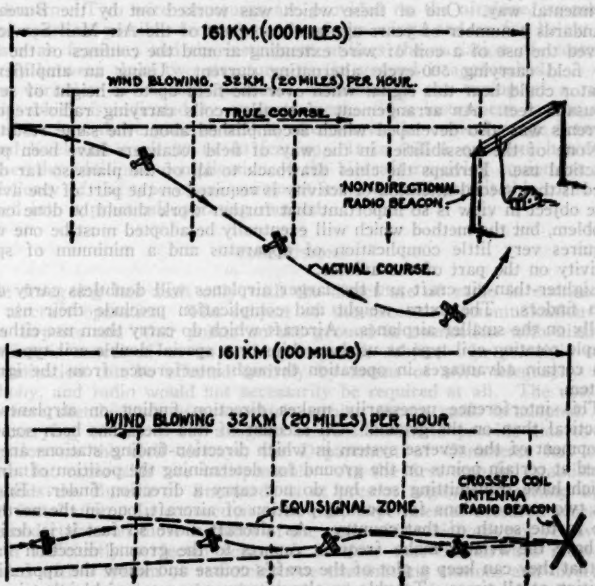
161 KM. (100 MILES)
WIND BLOWING. 32 KM. (20 MILES) PER HOUR.
TRUE COURSE.
NON DIRECTIONAL RADIO BEACON
ACTUAL COURSE.
161 KM (100 MILES)
WIND BLOWING 32 KM (20 MILES) PER HOUR
EQUISIGNAL ZONE.
CROSSED COIL ANTENNA RADIO BEACON

FIGURE 4.—COMPARISON OF EFFECT OF WIND DRIFT IN NAVIGATION BY DIRECTION FINDER (UPPER DIAGRAM) AND BY DIRECTIVE BEACON (LOWER DIAGRAM).

While the radio beacon will adequately direct flight along the course, it does not in its present form provide a positive means of determining the exact location of the landing field or of landing in fog. It serves in a general way as a field localizer since the width of the equisignal zone grows considerably narrower as the field is approached. It is quite possible, however, that it can be developed or supplemented so that the character of the signals or of the equisignal zone shall be particularized over the landing field and the exact location of the field definitely marked out. It is quite possible that further work on antenna systems for the beacon will accomplish this desirable purpose. A satisfactory field localizer would assist the aviator to a position directly above the field and direct him to a safe landing.

A method now used, and one which is capable of further development, is the simple one of having an observer on the landing field listen for the noise of the airplane and estimate its height above ground by the noise, sending this information by radio to the pilot. This involves utilizing the services of the ground personnel so that they can serve as the eyes or ears of the pilot for landing purposes. This same procedure may perhaps be used to aid in keeping the aircraft on a route, through listening at intermediate stations and indicating to the flier that he is passing a definite point or field on the route.

Several types of electrical field localizers have been developed in an experimental way. One of these which was worked out by the Bureau of Standards a number of years ago at the request of the Air Mail Service involved the use of a coil of wire extending around the confines of the landing field carrying 500-cycle alternating current. Using an amplifier, an aviator could hear this signal when over the field up to a height of several thousand feet. An arrangement of smaller coils carrying radio-frequency currents was also developed which accomplished about the same result.

None of the possibilities in the way of field localizers have been put in practical use. Perhaps the chief drawback to all of the plans so far developed is that special attention or activity is required on the part of the aviator. The object in view is so important that further work should be done on this problem, but the method which will eventually be adopted must be one which requires very little complication of apparatus and a minimum of special activity on the part of the aviator.

Lighter-than-air craft and the larger airplanes will doubtless carry direction finders. The extra weight and complication preclude their use generally on the smaller airplanes. Aircraft which do carry them use either the simple rotating-coil type as used on ships or a special double-coil type which has certain advantages in operation through interference from the ignition system.

This interference necessarily makes direction finding on airplanes less practical than on the ground. On account of this there has been some development of the reverse system in which direction-finding stations are provided at certain points on the ground for determining the position of aircraft which have transmitting sets but do not carry a direction finder. England has two such stations for locating position of aircraft, one in the north and one in the south of that country. As aircraft move so fast it is desirable to have the aviator make frequent reports to the ground direction finders so that they can keep a plot of the craft's course and know the approximate position at all times. Trouble on aircraft can occur so quickly that in some cases the flier would only be able to report that he was landing; there would be no time for a bearing, and it would be necessary to send aid or search in accordance with the plotted course. In some cases the reports would merely cease, and failure to arrive in due time would make the plot a valuable reference point.

The use of direction finders is an alternative to the directive-beacon system previously described. Aircraft which travel only fixed courses marked out by radio beacons would have no use for direction finders. Those which fly over courses not provided with the beacons, however, will find radio direction finding a valuable aid.

Work with aids to airplane navigation has first, last, and all the time the problem of landing as its chief difficulty. Much thought and experiment have been spent on methods to facilitate landing in fog, but the author finds that aviators declare that no mechanical instruments could ever give them sufficient confidence to make an actual landing when the ground was in-

visible. The radio beacon can guide a pilot to the landing field, and its location and outline can be given him by a field localizer device, but there still remains the problem of indicating to him the exact distance above ground during the last few feet of descent and actual landing. Various methods have been proposed or tried to give an indication of the exact distance above ground during this critical period when the airplane is settling to earth. All the methods offer great difficulties. They include electrical-capacity variation, sound reflection, and radio-wave reflection. The first of these is known as the capacity altimeter. It was developed by the Bureau of Standards and the Air Service in 1922, and is as promising as anything now known. Two wire structures are provided on the airplane, supported as far apart as possible from the ends of the wings and as far below the airplane as feasible; and the instrument utilizes the variation of electrical capacity between these two structures as the ground is approached.

The variation of capacity produces variations in the tuning of a radio circuit, which is indicated by an instrument. This instrument can be calibrated in feet above ground, and is effective in the critical range between ground level and some 30 to 40 feet up. It is still a question, however, whether the additional complication of the aircraft receiving equipment and the delicacy of the type of measurement involved will permit this to be developed into a practical operating method.

## CONCLUSION.

The foregoing has dealt only with the direct use of radio in connection with aircraft operation. The subject of the necessary communication between airports and other points on the ground in connection with aircraft operation is conceived as an entirely separate subject. In principle such communication can be effected entirely by means of wire telegraphy and telephony, and radio would not necessarily be required at all. The uses of radio here sketched are uses to which nothing except radio is adapted.

In carrying out its newly assigned responsibilities to provide aids to air navigation on the civil airways, the Department of Commerce has concluded that radio aids are indispensable. As the first step in establishing these aids the Bureau of Standards is at present engaged in setting up a model installation. In connection with this, experimental research work is being undertaken on those features of the radio aids which are not yet available in sufficiently perfected form. It is clear that the airways must be provided with a system of radiotelephone transmitting stations and directive beacons at certain intervals. At smaller intervals between the directive beacons, probably every 25 miles, are to be located the marker beacons. It is not yet certain whether the beacons will operate by means of an audible or a visual signal on the airplane, and the determination of this is one of the principal objects of the investigation now in progress.

It would be impossible to prophesy the limit of possibilities of radio in connection with air navigation. In addition to radiotelephony, the directive beacon, and the marker beacon, there remain the further possibilities of field localizers, direction finders, and landing altimeters. Research is in progress on these additional aids. They all offer difficulties, but will doubtless be eventually worked out and adapted in various forms to aircraft use. In any event, there is every reason to believe that radio will have a steadily increasing part in expediting, and increasing the safety of, air navigation.—"Mechanical Engineering," January, 1927.

## CENTRIFUGAL CASTING AND ITS PROBLEMS.*

The development of the centrifugal casting process and its application to the high melting-point alloys have given rise to a series of new problems in metallurgy and the testing of materials. The investigations that have been made up to the present time on foundry practice apply in the largest part to the sand casting process, so that the results obtained from them can be applied only in the most limited manner to the centrifugal casting process. These two processes are essentially very different. These characteristic differences will be better understod by examining the technical details of the two processes.

*Sand and Centrifugal Processes Compared.*—In the sand casting process a large number of moulds are employed, and a definite weight of metal is melted and poured into these moulds and allowed to congeal therein. In the centrifugal casting process a large number of castings are made in a single steel mould, the process being continuous. Hence a large quantity of metal is maintained in the molten state, and sufficient of the molten mass is run off each time to make the casting, while new metal is added to the melting-pot from time to time. The molten metal is pressed into the mould by a suitable device, and congelation takes place immediately, whereat the still hot casting is ejected and the mould is ready for the next casting.

The differences between the two processes are easy to recognize and need not be gone into in great detail. In the sand casting process the duration of melting and the mass of metal melted each time are magnitudes which are accurately known, and therefore it is comparatively easy to determine beforehand the course of the reactions that take place in the melting process. It is also possible to counteract these reactions, when they are undesirable, by adding the proper substances to the metal bath. Thus directly before the metal is poured it is possible to add reagents which rapidly react to remove impurities.

On the other hand, in the centrifugal casting process the chemical condition of the metal bath is constantly altered by agencies which act continuously on the molten metal, such as the air, the walls of the crucibles, or container in the pressure chamber. In this case the conditions are entirely different, for it has been found that such impurities, as, for example, the gases that are absorbed by the molten metal, when once they are contained in it, have no opportunity of escaping after the casting process, because the metal solidifies almost immediately in the centrifugal form. Thus gases in the molten metal are present as air-pockets in the solid metal. Hence it is easy to see that the treatment and handling of the metal are matters of considerably more difficulty in the centrifugal casting process than in the sand casting process.

*Alloys Better for Centrifugal Castings.*—The solution of the problem lies in the realm of alloying. Inasmuch as, when once an impurity or an imperfection in the molten-metal bath occurs in the centrifugal casting process, it remains—for it can be removed only to a very limited degree—it follows that alloys must be selected for use in the centrifugal casting process which are peculiarly adapted to it, and which are resistant to the various injurious chemical agencies that are present during the process. Thus these molten alloys must possess the property of not absorbing gases, of not holding fast particles of slag and the like in their interior, and of reacting but very little with the air that is found in the centrifugal mould.

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* From an article by L. Frommer in "Zeitschrift für Metallkunde," 1925. Translated in the "Iron Age."

These are rather difficult conditions, but there have been found a number of alloys which can be cast in a practical manner by this process, and the castings that are obtained are satisfactory from the physical standpoint.

*Phenomena in the Mould.*—The phenomena which take place in the mould after casting form an even greater contrast between the sand casting process and the centrifugal casting process. The sand form is made in such a manner that it offers no resistance to speak of to the contraction that takes place when the molten metal congeals in it. The heat conductivity of the substance from which the sand form is made is very slight. The moulds are also prepared in the proper manner to allow gradual cooling to take place.

Hence, due to the properties of the sand mould as explained above, in most cases it is possible to allow the metal, after it has been poured into the mould, to remain there until it has entirely cooled off. The cooling takes place slowly. The alloy metals that are cast in this manner are thus given a chance to reach a condition of equilibrium, and the strains that take place during the contraction of the casting are not induced by the resistance of the mould to the shrinkage, but only to the unequal rate of cooling that may occur in certain parts of the cast metal. Only in the case of very large and very irregular castings is it found that the strains induced in the solidification of the metal are so large that there is danger of the casting being disrupted when subjected to pressure.

When considering the shrinkage of the metal during cooling, the important matter is not the contraction that may take place while the metal is cooling, but the total shrinkage that occurs from the time the metal is poured at the high temperature to the time that it attains room temperature. This maximum shrinkage in the case of the sand casting process amounts to only a very small fraction of the dimensions of the finished casting.

On the other hand, and in direct contrast to these conditions, the progress of the solidification of the molten metal and the cooling process are of considerable importance in the centrifugal casting process. The illustration shows a simple centrifugal casting mould as well as the finished casting itself. Thus in this *v* is the top half of the mould, *h* the bottom half, *e* is the opening through which the metal enters, and *g* is the casting. The core is seen at *k*. The studs *a* are for removing the casting after it has solidified.

As soon as the metal has filled the mould it quickly solidifies and cools off very rapidly. The changes in condition that take place during the congealing of the metal are generally suppressed, so that the alloy metal which is cast by this process and in which these changes occur is not in a state of equilibrium after cooling is completed. It is of still greater importance that the steel mould offers a certain amount of resistance to the contraction of the molten metal during the cooling process. As long as the casting is in the mould it can contract only in that direction in which the steel mould is not fixed. Hence the casting which is shown can shrink only in the direction of E F, while every tendency to contract in the directions G H and J K is prevented, and only the thickness of the walls of the casting can be reduced. Hence it follows that stresses are induced in the casting in that direction. It is only when the metal has cooled down sufficiently so that it can be removed from the mould that contraction can take place unhindered.

*Conditions for Avoiding Cracks.*—The suitability of a metal for casting by the centrifugal process depends on whether or not it can remain in the mould until it attains such temperatures that it can be expelled therefrom without being subjected to the formation of cracks due to contraction. In addition to the first condition there is also a second of equal importance. A casting as shown in the illustration contracts while it is cooling, and a cer-

tain pressure is exerted against the core *k* and the posterior portion of the mould *h*. The result is that when the casting is pulled out of the mould, stresses are induced in it along the axis C D, and when the core is pulled out along the axis A B, as may be easily seen. These induced strains are apparently greater the more the casting has cooled off up to the time that it is ejected from the mould.

On the other hand, the mechanical strength of the casting increases as the temperature is reduced. Thus, if the casting is to be brought out of the mould in good condition, it is essential that the alloy attain such properties during the cooling period within the mould that the tension due to shrinkage to which it is subjected is small, while its mechanical strength is large. Similarly, the hardness of the casting should have reached such a point by the time it is ready for removal that the rods which are used to aid in its removal from the mould do not penetrate into the metal.

In order to establish the suitability of the metal for casting by the centrifugal process, the first test that must be made on it is to determine whether it will satisfy the two aforementioned conditions. It is a fact that metallurgical investigations have not afforded much information on this particular subject, and such information as is available is limited to the determination of the specific mechanical strength of the casting at the temperatures at which it is employed. Of far greater importance is the variation of the contraction, the strength and the elongation of the casting as the cooling process progresses, and the relationship that exists between these characteristics of the casting and the temperature. This is the main subject-matter of what follows.

In the original article a discussion is given of the magnitude and distribution of casting strains, and of the importance of these factors in determining the tendency towards the formation of fissures and cracks in castings made by the centrifugal method. It is shown that this tendency of a casting depends on the behavior of the metal from the time that it flows into the mould to the time that it is ejected from the mould; or, speaking in terms of temperature, from the temperature at which it enters the mould to the temperature a which it is removed from it.

*Testing the Centrifugal Casting.*—There are two tasks that must be performed by the methods that are employed in the testing of centrifugal castings. In the first place, it is evident from what has preceded that the raw materials must first be tested to determine whether or not they are suitable for being cast by the centrifugal process, and the degree of this suitability must be determined quantitatively. Then, in the second place, it is necessary to determine the mechanical properties of the alloys which are found to be usable in this manner, and this must be accomplished in such a manner that the influences which affect the mechanical properties of the casting during the processes of casting, quenching, and cooling are taken into consideration.

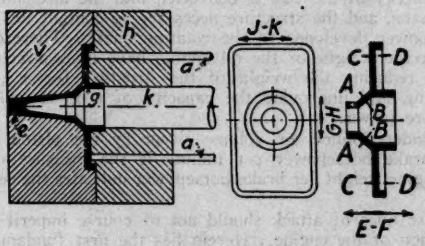
It is only after these methods have been developed that it will be possible to establish the value of the centrifugal casting process for this purpose, and also to select the materials that are to be handled in this manner in such a way that the finished products will be perfectly suited to the purposes for which they are intended.

In testing the suitability of alloys and metals for the centrifugal casting process it is necessary to determine the behavior of the metals in the warm condition. The course of the shrinkage curve as the metal changes in temperature must be established. When this is done it will be possible to obtain an accurate idea of the manner in which the alloy will behave while it is being centrifugally cast. Then, again, it is necessary to determine the flexibility of the metal and the tension under conditions of shock.

An easy method of establishing the manner in which the metal behaves while it is cooling off in the casting process is to cast a test piece in such form that both ends of the test rod can be maintained rigid during the testing process. In this way it may be determined whether or not cracks will form in the casting while it is cooling off.

It is not possible to give all the details of this portion of the investigation in this abstract, and the reader is referred to the original for them.

The second part of the testing problem is to determine the properties of the finished casting in the cold. The testing of a casting made by the centrifugal process is somewhat different from that made by the sand process. There are certain changes which the metal undergoes in the centrifugal mould and certain influences to which it is subjected that render it necessary to make certain special tests on it.




CENTRIFUGAL CASTING.

The porosity of the casting must be taken into consideration. The variation in the density of the metal in the centrifugal casting on the inside as compared with the surface is important. The metal on the surface is more dense than in the interior of the casting. Then, again, the influence of the irregularity of the size of the core must be considered. The quick cooling at the walls of the form has the result that the limiting zones of the casting contain very fine grains of metal, while the granulation becomes coarser as the interior of the casting is approached. Finally, there must be considered the fact that the metal is cast in the mould under pressure, and this is bound to have some effect on the mechanical properties of the casting.

It is therefore understandable that, when the metal near the surface of the casting is tested for strength, ductility, and the like, different values will be obtained from those when metal in the interior of the casting is tested. These facts must be carefully considered in determining the strength of a casting made in this manner, and in establishing the suitability of the centrifugally cast object for one purpose or another. At best the problem is one of great complexity, and while there are many advantages to be obtrained from the centrifugal casting process, it is well to consider all phases of the problem before deciding that the centrifugal process is the one to employ.—"Mechanical World," November 12, 1926.

## THE MODERN OIL ENGINE.

BY EDWARD C. MAGDEBURGER,[1] WASHINGTON, D. C.

Despite the existence of a large number of highly instructive technical papers dealing with various phases in the progress of the oil engine toward

[1] Aide on Diesel Engines, Bureau of Engineering, Navy Department.

its ultimate place in modern life, and probably for that very reason, a brief review of the efforts of the small army of engineers engaged in oil-engine building would be of interest to fellow engineers pushing other wheels of civilization. The ability of the Diesel type of oil-engine to burn low grades of oil, and with great efficiency, has been so thoroughly advertised that an immediate demand exists for oil engines suitable for propulsion of warships, aircraft, locomotives, trucks, and automobiles. The failure of the existing commercial types to qualify for these services is largely due to their inability to meet weight and space requirements, although engines ranging in size from 1 horsepower to 2000 horsepower per cylinder have already been constructed and tested. A unit of 15,000 horsepower in nine cylinders has been installed and put into operation.

An oil-engine unit consists essentially of a combustion chamber where the chemical energy of the fuel is converted into the mechanical energy of the moving piston, and the structure necessary to contain this chamber and transmit the power developed to the rotating shaft. Therefore efforts to reduce the specific weight of the oil engine have been directed along two distinct lines: reducing the weight of the structure for a given effective cylinder content, and improving the capacity of a given cylinder volume to develop more power.

Specific cylinder volume, or volume swept by the piston, necessary to develop one brake horsepower per minute is the measure of the latter, whereas the engine weight per brake horsepower indicates the combined effect of both.

Both of these lines of attack should not of course imperil the safe and reliable operation of the engine. Herein lies the first fundamental division of engineering opinion. Some see in the two-stroke cycle, giving roughly twice the horsepower out of the same cylinder contents, a feature worth developing. Others assert that the established and unquestioned reliability of the four-stroke cycle must be maintained, and other means found to offset the disadvantage of its requiring twice the time for the same amount of work developed. Much heated debate resulted in the past over this point and it is not the intention of the author to champion the cause of either side but rather to attempt to record the actual efforts put forth by both to prove their point.

## I—FOUR-CYCLE ENGINES.

Oil engines of the four-cycle type, being historically the first to be developed commercially, have attained a very high degree of reliability, and show only minor variations in the design of the vital parts. Much operating experience has been gained in various fields of application extending over long periods of time, and with widely different types, and the line of progress is very much easier to determine.

One of the greatest hopes for development of the four-cycle engine lies in the adoption of higher shaft and piston speeds, largely brought about by the splendid performance of the German submarine engines during the late war. The largest of these engines, which has, with minor modifications and slight reduction in rating, been adopted as a part of the propelling plant of some of the new U. S. submarines (Fig. 1), stands out as the utmost dared in this direction by the designer (No. 1, Table 1). The war was over before any service experience was obtained with these engines, and,

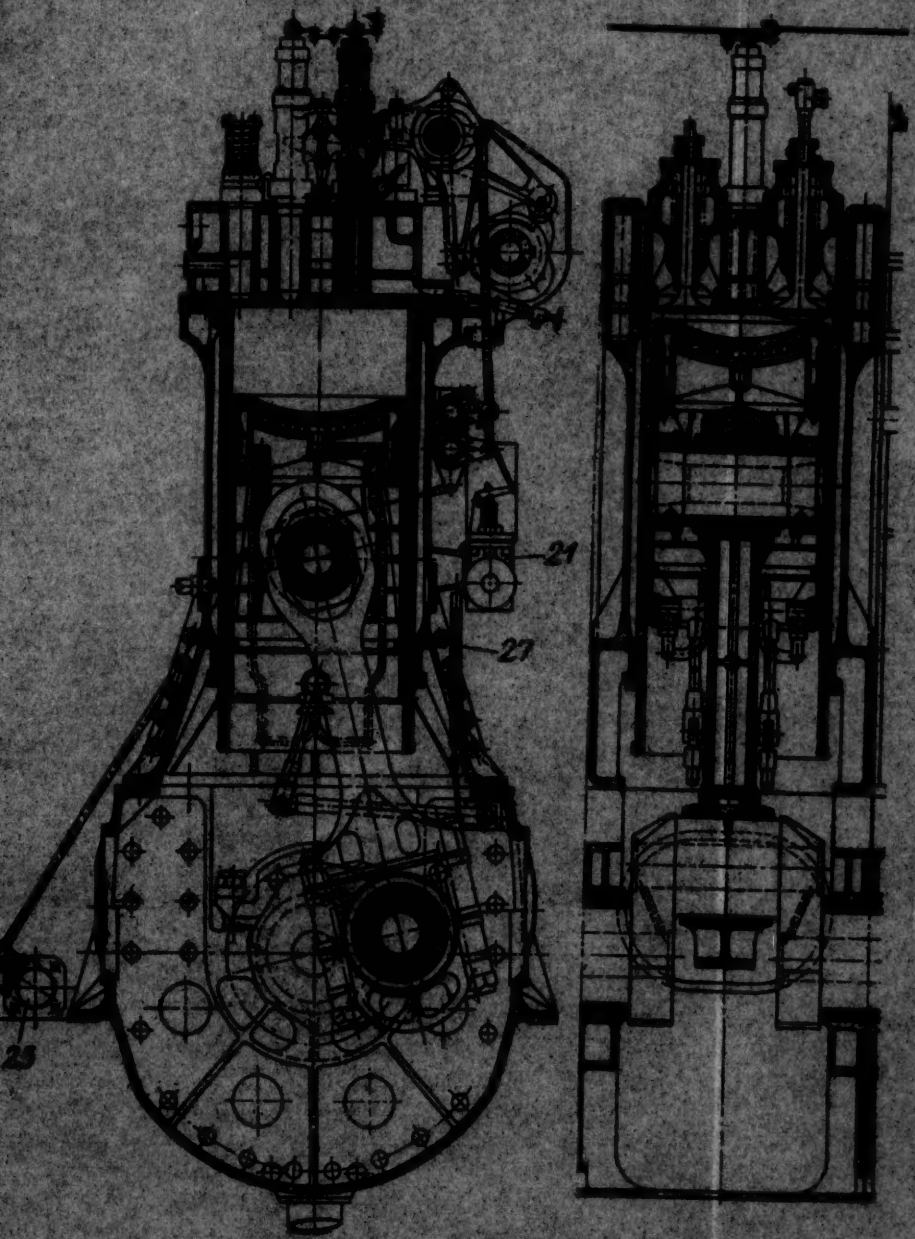



FIGURE 1.—SECTIONS THROUGH CYLINDERS OF THE 3000-B.H.P. GERMAN SUBMARINE ENGINE (M.A.N.)

## TABLE 1 DATA ON FOUR-CYCLE ENGINES[1]

| No. | Type of engine | S.A. or D.A. | No. of cylinders | B.Hp. Total | B.Hp. One cylinder | Bore, in. | Stroke, in. | R.p.m. | Piston speed ft. per min. | M.E.P. lb. per sq. in. | Specific volume, cu. ft. per hp-min. | Weight, lb. Total | Weight, lb. Per b.hp. | Remarks |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | German submarine | S.A. | 10 | 3000 | 300 | 20.86 | 20.86 | 390 | 1356 | 85.4 | 5.35 | 207,000 | 69.0 | Submarine rating |
| | | | | 1650 | 165 | 20.86 | 20.86 | 230 | 800 | 79.6 | 5.74 | 207,000 | 125.0 | Motorship rating |
| 2 | Burmeister & Wain | D.A. | 6 | 6750 | 1125 | 33.07 | 59.05 | 125 | 1238 | 73.5 | 6.23 | 1,180,000 | 168.0 | Without air compressor |
| 3 | Cramps-B. & W. | S.A. | 6 | 1700 | 283 | 29.13 | 45.28 | 115 | 867 | 64.8 | 7.09 | 600,000 | 352.0 | |
| 4 | Maybach | S.A. | 6 | 140 | 23 | ..... | ..... | 1300 | ..... | ..... | ..... | 2650 | 18.9 | |
| 5 | Nobel (1908) | S.A. | 8 | 200 | 25 | 7.87 | 8.66 | 600 | 865 | 78.4 | 5.84 | 4400 | 22.0 | V-type |
| 6 | McIntosh & Seymour | S.A. | 12 | 1000 | 83 | 14.0 | 18.0 | 325 | 975 | 73.3 | 6.25 | 80,000 | 80.0 | V-type |
| 7 | Hamburg-Mannheim | S.A. | 6 | 1600 | 267 | 27.16 | 39.37 | 120 | 787 | 77.5 | 5.92 | 385,000 | 240.0 | |
| 8 | N.-E. Werkspoor | D.A. | 6 | 4000 | 667 | 32.28 | 59.05 | 95 | 935 | 60.6 | 7.56 | 1,165,000 | 291.0 | |
| 9 | Pacific-Werkspoor | S.A. | 8 | 2900 | 363 | 31.5 | 51.18 | 95 | 810 | 76.0 | 6.04 | 1,100,000 | 380.0 | |
| 10 | McIntosh & Seymour | D.A. | 4 | 2700 | 675 | 32.0 | 52.0 | 95 | 823 | 70.9 | 6.46 | 760,000 | 281.5 | |
| 11 | McIntosh & Seymour | S.A. | 6 | 2700 | 450 | 32.0 | 60.00 | 95 | 950 | 77.8 | 5.9 | 800,000 | 296.0 | |
| 12 | Beardmore-Tosi | D.A. | 8 | 800 | 267 | 20.0 | 24.5 | 240 | 980 | 60.3 | 7.61 | 157,000 | 196.0 | |
| 13 | Beardmore-High Speed | S.A. | 8 | 400 | 50 | 8.25 | 12.0 | 750 | 1500 | 82.3 | 5.56 | 4500 | 11.25 | |
| 14 | Blohm & Voss-M.A.N. | S.A. | 6 | 1750 | 292 | 23.62 | 27.56 | 215 | 988 | 89.1 | 5.15 | 363,000 | 209.0 | Supercharged |
| 15 | Italian-B. & W. | D.A. | 8 | 10,000 | 1250 | 33.07 | 59.05 | 125 | 1238 | 81.7 | 5.6 | 1,900,000 | 190.0 | Supercharged without air compressor |
| 16 | Harland & Wolff-B. & W. | D.A. | 8 | 7500 | 938 | 33.07 | 59.05 | 115 | 1131 | 67.2 | 6.82 | ..... | ..... | With air compressor |

[1] Ten per cent allowance made for piston-rod area.

although a number of them have since been installed in merchant ships,[2] their rating has been considerably reduced, as shown by the lower figures in Table 1.

### HIGHER SPEEDS TO REDUCE WEIGHT AND SPACE REQUIREMENTS.

However, as the result of this daring, the well-known Burmeister and Wain Company built its double-acting experimental cylinder, which has been since incorporated in a number of large engines in apparently successful operation on the high seas. The high piston speed of this engine (No. 2, Table 1), if justified by service experience,[3] is a considerable advance over the prevailing practice of the same builder, and contributed considerably to the reduction in weight shown as compared with single-acting engines built by its American licensees—Cramps—(No. 3, Table 1). It is true that another factor, namely, introduction of the double-acting principle and the omission of the injection-air compressor (both of which will be discussed later) contributed largely to the result, but the fact that, the specific weight has been reduced by $(1238—867)/1238 = 30$ per cent due to speed increase alone, can hardly be denied.

A notable example, where the designer achieved considerable advance in the direction of weight and space reduction by relying only on the use of high relative speed, is the Maybach[4] engine (Fig. 3), said to be rather extensively used in Germany in rail cars and small marine applications (No. 4, Table 1). Here also the orthodox air injection of fuel has been retained, although the modern tendency is to begin lightening up the engine with the elimination of the injection-air compressor.

*Intermediate Transmissions.* The question arises, naturally, what is it that stands in the way of progress, by increasing both the piston and rotative speeds. There are two types of difficulties encountered: one set of troubles that will have to be overcome is mechanical, the other thermodynamical. To the mechanical group belong the necessity of using intermediate transmissions, such as gear drives,[4,5] electric transmission,[6] and hydraulic type of gear,[7] all of which introduce additional weight to offset the gain from the high speed of the oil engine itself, and additional complication of installation, which may be quite considerable and may add a questionable reliability factor of the transmission itself. Proponents of these drives,[8] possessing quite naturally a better knowledge of the devices, do not admit the

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[2] *Munsterland*, *Havelland*, and *Vogtland*. See "The Motorship" (British), Oct., 1928, p. 225; and "Motorship" (N. Y.), May, 1928, p. 335.

Contributed by the Oil and Gas Power Division for presentation at the Annual Meeting, New York, December 6 to 9, 1926, of the American Society of Mechanical Engineers. All papers are subject to revision.

[3] M. S. *Gripsholm*. See "Motorship" (N. Y.), Dec., 1925; "Mechanical Engineering," Jan., 1926, p. 81. It is hoped that engines built by the British licensee of Burmeister & Wain Co., of exactly the same cylinder dimensions but running at 115 r.p.m. (M. S. *Asturias*—see No. 16, Table 1) and 96 r.p.m. (M. S. *Carnarvon Castle*, "The Motorship" (British), July, 1926, p. 120), do not reflect service experience with the high piston speed of engines in the M. S. *Gripsholm*.

[4] "Oil Engine Power," Nov., 1924, p. 595; Dec., 1924, p. 658.

[5] A considerable number of gear-drive installations for tugs have been made in this country. See "Motorship" (N. Y.), Nov., 1925, p. 826; Aug., 1926, p. 590; July, 1926, p. 521, but the U. S. Shipping Board refused to recognize this solution for ship propulsion, although it was the cheapest offered.

[6] Electric drive has recently been successfully applied in considerable number to tugboats (See "Motorship" (N. Y.), Feb., 1925, p. 115, tow boats; July, 1925, p. 505, ferries; Aug., 1926, p. 599, Sept., 1926, p. 674, tankers; and May, 1926, p. 352, locomotives.

[7] Vulcan Hydraulic Gear for Ships, "Motorship" (N. Y.), Jan., 1925, p. 28.

[8] Dr. Frahm, of Blohm-Voss Co., has contributed a most interesting paper comparing weight and space requirements of high-speed geared installations for ship propulsion. See "Motorship" (N. Y.), Feb., 1925, p. 103.

existence of any such questionable features, and the service experience gained with the pioneer installations apparently vindicates their convictions since new installations of either of the three indirect drives for ships and locomotives are being made extensively.

*Destructive Critical Speeds.* Another bugaboo of the mechanical group that is being rapidly put under control is the destructive critical speeds resulting from synchronous torsional vibration. It is when the high-speed oil engine for submarine service came into existence that these were encountered, and to the "dreadful" submarine must go the credit for eliminating torsional vibration from the path of oil-engine progress. With the increase in engine speed also increases the number of torque impulses conveyed to the shaft, and since a given shaft system has a constant frequency of free or natural vibration, the periods of synchronism descend into the operating range and may subject the shafting to excessive stresses resulting in failure. This trouble, however, has been largely eliminated, and although improvement in the methods used is still desirable, no destructive critical torsional speeds need exist through the operating range, unless the required range is excessively large as in propelling plants of submarines, or other warships for that matter, where such a long engine operating range, as in one instance from 75 to 450 R.P.M., is very desirable, and where it must be unavoidably somewhat restricted.

*Increased Heat Stresses.* Another type of trouble that had to be conquered was the increased heat stresses resulting from increase in engine speed. These are confined mostly to piston crown and cylinder head, and with the existence of the two idle strokes of the four-stroke cycle do not reach excessive limits and become of importance only after the cylinder bore exceeds 10–12 inches in diameter. At any rate, provision for free expansion of both piston and cylinder head in the B. & W. design (Fig. 2), or the use of high velocity of cooling medium in the M.A.N. design (Fig. 1) of cylinder head and piston, solves the problem.

Another part that is exposed to the increased heat stresses is the exhaust valve. Here the new heat-resisting alloy steels solve the problem for smaller sizes in the same way as they did for the high-speed automobile and aircraft motors. Internal cooling of the valve seat and stem used on very large engines adds but very little to the unavoidable complication of the water-cooled valve cage. The cylinder-head design introduced by Tosi (Fig. 4) provides a different and very attractive solution of the heat problem by using smaller valves in the head, permitting better cooling of the head, but using them both for inlet and exhaust, so that the incoming induction air serves to remove the heat absorbed by the valves from the hot exhaust during the immediately preceding exhaust stroke. Such double-duty valves have been adopted also in the new design of Werkspoor double-acting engine (Fig. 7).

Besides the increase in speed, modifications in the orthodox means (requiring a crank mechanism for each cylinder) to transmit the power developed to the shaft have been studied, with the result that several solutions have passed the inevitable criticism following the attack of anything as established as was the single-acting four-cycle type. Many more have been proposed from time to time, but the courage of their sponsors has so far not been sufficient to overcome the obstacles on their way to adoption by oil-engine builders.

*V-Type Construction.* As early as 1908, when Nobel, the oil magnate of Russia, desired a propelling plant for his yacht, the weight and space requirements could not be met by the four-cycle engine of the day in any other

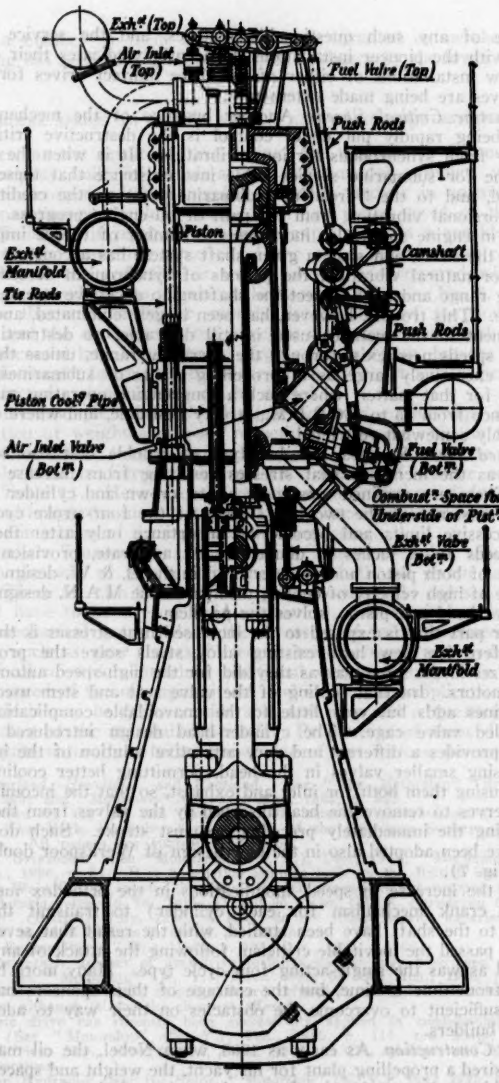



FIGURE 2.—CYLINDER SECTION OF BURMEISTER & WAIN DOUBLE-ACTING ENGINE.

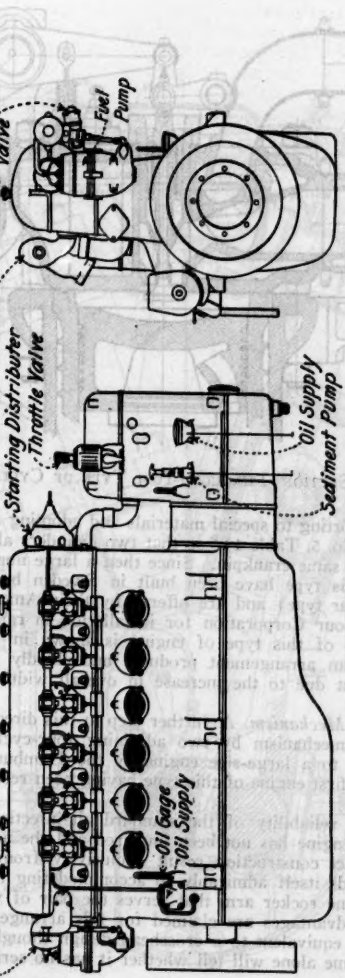



FIGURE 3.—MAYBACH OIL ENGINE.

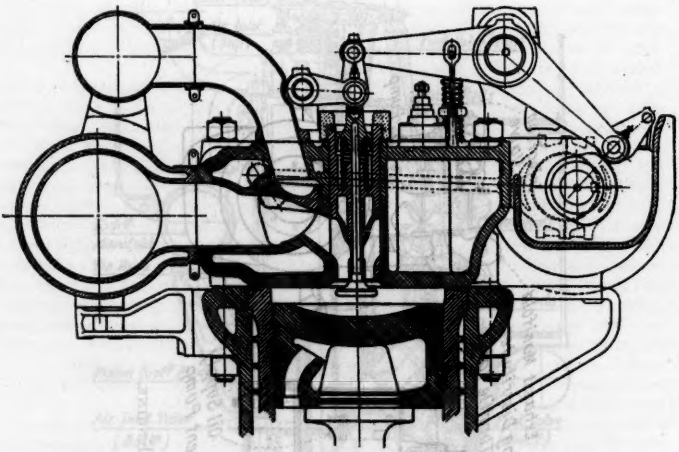


FIGURE 4.—SECTION THROUGH TOSI TYPE OF CYLINDER HEAD.

way except by resorting to special materials and adopting a so-called V-type of construction (No. 5, Table 1),[9] so that two cylinders alternately transmit their torque to the same crankpin. Since then a large number of moderate-size engines of this type have been built in Sweden by the Atlas Diesel Aktiebolaget (Polar type) and are offered by their American licensee, the McIntosh & Seymour Corporation for installation in railroad cars.[10] The latest development of this type of engine is shown in Figure 5 (No. 6, Table 1). Such an arrangement produces undoubtedly a more light and compact engine but due to the increase in overall width is unsuitable for many applications.

*Common Crank Mechanism.* A further step in this direction is the use of a common crank mechanism by two adjoining four-cycle cylinders alternately, as adapted to a large-size engine by the Hamburg-Mannheim Co., Ltd. (Fig. 6), the first engine of this type having been recently installed in a motorship.

Here again the reliability of the standard air-injection type of single-acting four-cycle engine has not been impaired and the advantages of light weight and compact construction result exclusively from the arrangement adopted which lends itself admirably to accommodating the air-compressor drive from the same rocker arm that serves the pair of working cylinders. Many additional advantages are claimed for this arrangement such as, for instance, that it is equivalent to a crosshead design though much shorter in height, etc.; but time alone will tell whether it has no serious disadvantages of its own.

*The Double-Acting Principle.* Among the recent arrivals into the oil-engine industry is the application of the double-acting principle. This statement needs probably to be modified slightly since four-cycle double-acting

[9] Trans. Inst. Eng. and Shipbuilders in Scotland, Apr., 1926, p. 566.
[10] "Oil Engine Power," Nov., 1923, p. 529.

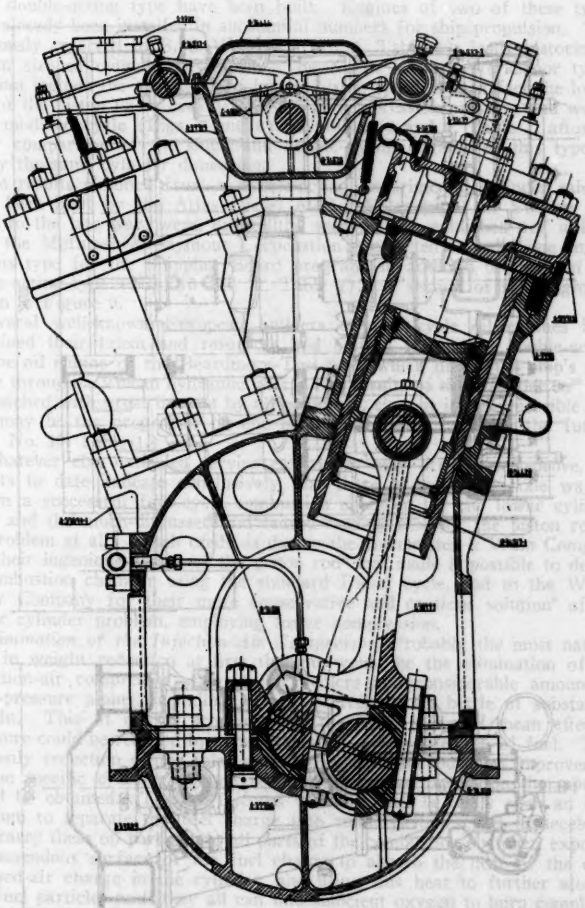


FIGURE 5.—SECTION THROUGH MCINTOSH & SEYMOUR LOCOMOTIVE ENGINE (1000 B.H.P.).

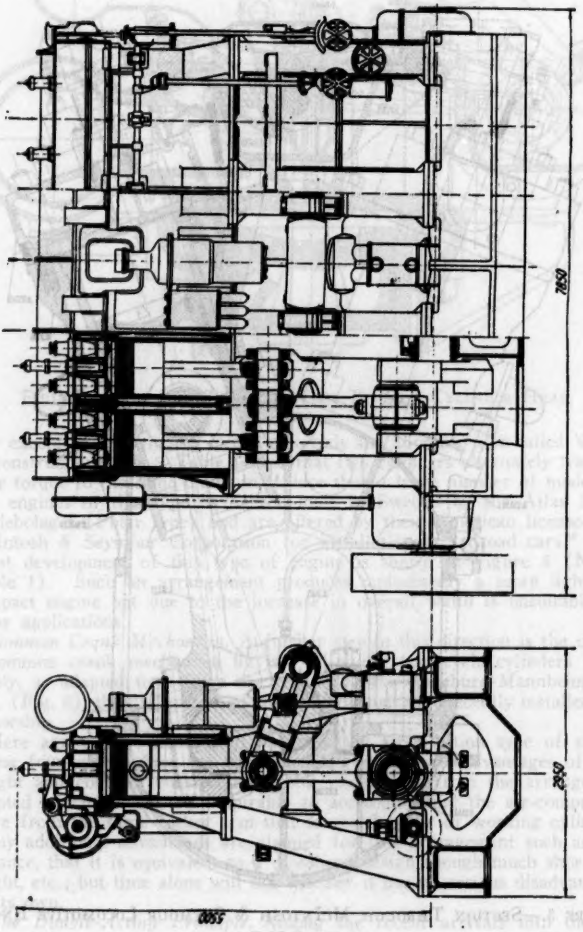



FIGURE 6.—1600-B.H.P. HAMBURG-MANNHEIM ENGINE.

oil engines of the horizontal tandem type were built by the M.A.N. Company years ago in considerable numbers. Reference is made here to the recent efforts of applying this principle to vertical engines, suitable for ship propulsion. As many as four different experimental cylinders of the four-cycle double-acting type have been built. Engines of two of these types have already been installed in substantial numbers for ship propulsion. The previously referred to B. & W. type[9] (No. 2, Table 1), leads historically and in size of cylinder developed. The North Eastern-Werkspoor type[11] does not include the advantage of higher piston speed, and in fact the lower side of the piston has a considerably lower compression pressure and works on a modified cycle (Figs. 7 and 8). Numbers 8 and 9, Table 1, afford a ready comparison of the single-and double-acting engines of this type of nearly the same cylinder dimension.

The results obtained from a double-acting experimental cylinder said to have been built by the Atlas Diesel A.-B. (Polar type) in Sweden just prior to the late war, were, due to the war, neither published nor utilized until the McIntosh & Seymour Corporation contracted to build one engine of this type for the Shipping Board program, in addition to three of the single-acting type (Nos. 10 and 11, Table 1). A section of this engine is shown in Figure 9.

Several well-known European builders of four-cycle oil engines have combined their talent and resources and built a high-speed double-acting marine oil engine of the Beardmore-Tosi type, which drives the ship's propeller through a Vulcan hydraulic gear. The results of this installation[12] will be watched with great interest by all engineers, since it is quite possible that this may be the progenitor of one of the oil-engine types of the future. (See No. 12, Table 1.)

Whatever else extended service experience may prove or disprove, the results to date indicate conclusively that there is more than one way to design a successful four-cycle combustion chamber for the lower cylinder half, and the much-discussed and feared stuffing box for the piston rod is no problem at all. Much credit is due to the Burmeister & Wain Company for their ingenious design of the piston rod[9] that made it possible to design a combustion chamber using the standard Diesel cycle, and to the Werkspoor Company for their more conservative and cautious solution[9] of the lower cylinder problem, employing lower compression.

*Elimination of the Injection-Air Compressor.* Probably the most natural step in weight reduction at first thought would be the elimination of the injection-air compressor with its air coolers and considerable amount of high-pressure piping, including an air receiver or air bottle of substantial weight. This of course provided that the same indicated mean effective pressure could be realized by some system of airless injection of fuel. Then not only reduction of the total weight of the engine but also improvement in the specific cylinder volume required to realize one brake horsepower could be obtained. The air injected with the fuel is really such an ideal medium to separate the fuel charge into small particles and to accelerate and carry these oil particles to all parts of the combustion chamber, exposing a tremendous surface of the fuel charge to absorb the heat of the compressed-air charge in the cylinder, and using this heat to further atomize the fuel particles until they all can find sufficient oxygen to burn completely and without producing smoke. Small wonder, then, that the first substitutes for air injection found but little appreciation of their rather mediocre per-

[11] "Motorship" (N. Y.), Feb. and July, 1926, pp. 124 and 526; "The Motorship" (British), June and Oct., 1926, pp. 84 and 241.
[12] "Motorship" (N. Y.), May, 1925, p. 370.

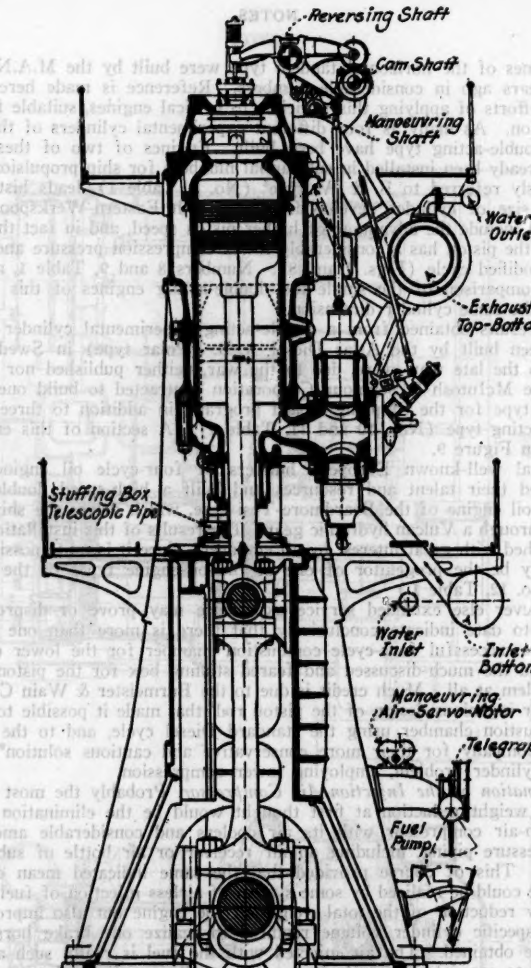



FIGURE 7.—CYLINDER SECTION OF WERKSPOOR DOUBLE-ACTING ENGINE (NEW DESIGN).

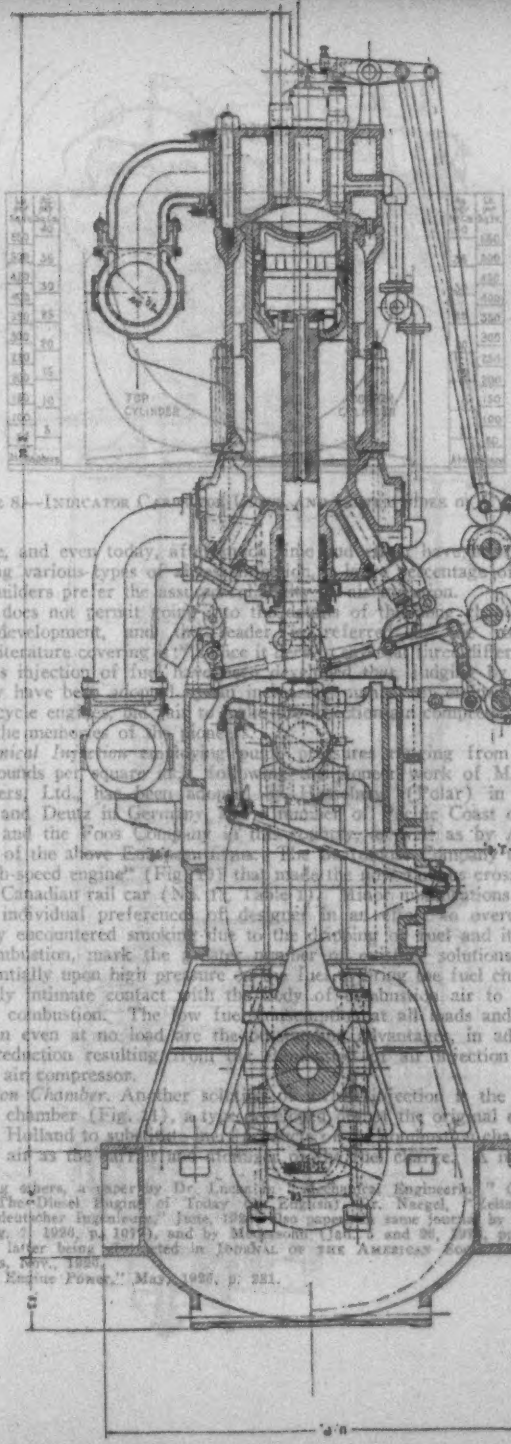



FIGURE 8.—INDICATOR C[illegible] OF THE [illegible] TYPES OF ENGINE.

formance, and even today, a[illegible] developing various types [illegible] engine builders prefer the [illegible] Space does not permit [illegible] engine development, [illegible] current literature covering [illegible] of airless injection of [illegible] that they have be[illegible] of four-cycle engi[illegible] only in the memo[illegible]

*Mechanical Injection.*—[illegible] ranging from 2000 to 20,000 pounds per [illegible] of McKechnie of Vickers, Ltd., [illegible] Polar) in Sweden, M.A.N. and Deutz in Germany, [illegible] Coast oil-engine builders and the Foos Company [illegible] as by American licensees of the above [illegible] their high-speed engine" (Fig. [illegible]) that made the [illegible] cross-country run in a Canadian rail car (N[illegible] ations embodying the individual preferences [illegible] to overcome the originally encountered smoking [illegible] fuel and its incomplete combustion, mark the [illegible] solutions, but all rely essentially upon high pressure [illegible] the fuel charge into sufficiently intimate contact [illegible] air to result in complete combustion. The low fue[illegible] all loads and the fine regulation even at no load are the [illegible] in addition to weight reduction resulting [illegible] jection and the attached air compressor.

*Injection Chamber.*—Another [illegible] jection is the so-called injection chamber (Fig. [illegible]), a type [illegible] the original efforts of Brons in Holland to sub[illegible] chamber for injection air as the [illegible] Chamber of

[illegible]

[illegible] Among others, a paper by Dr. [illegible] Engineering," Oct. 1921, p. 534; "The Diesel Engine of Today," [illegible] Nagel, "Jahrbuch des Vereines deutscher Ingenieure," June, 1[illegible] same journal by O. Eichelberg (Aug. 7, 1920, p. 1019), and by M[illegible] and 26, [illegible] 767 and 931), the latter being [illegible] JOURNAL OF THE AMERICAN [illegible] OF NAVAL ENGINEERS, Nov., 1920.

[illegible] "Oil Engine Power," May, 1922, p. 231.

FIGURE 9.—SECTION OF THE McINTOSH & SEYMOUR DOUBLE-ACTING ENGINE.

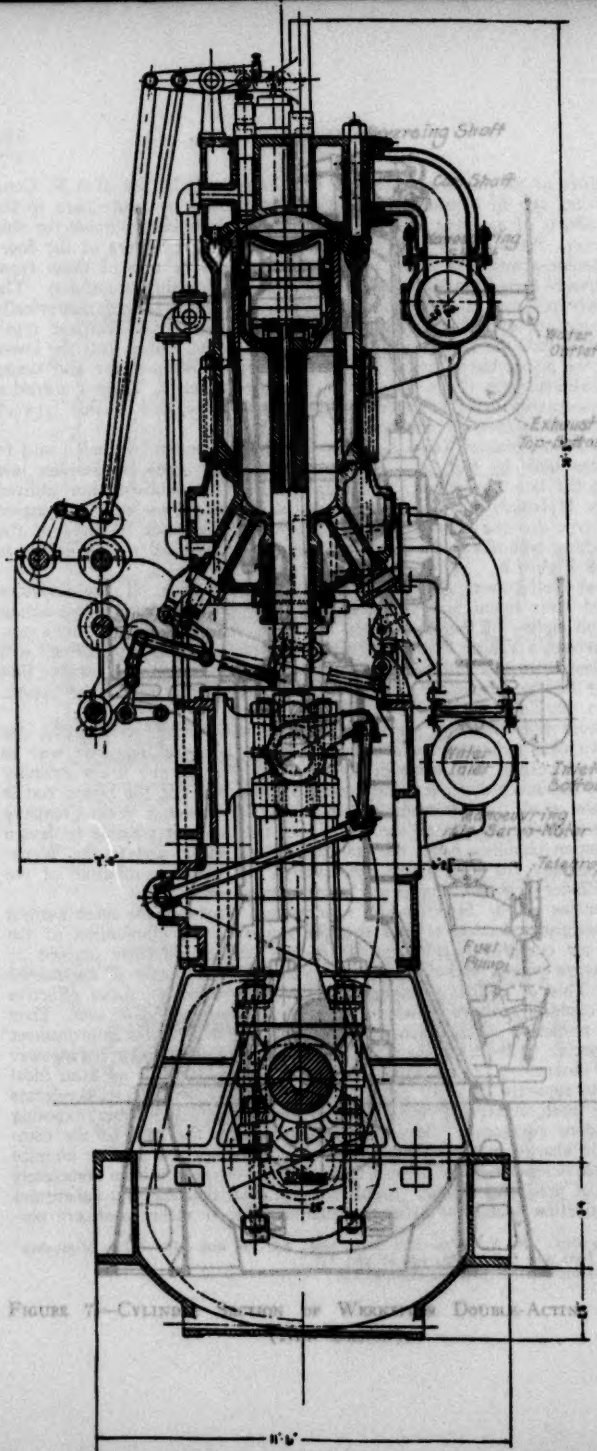



FIGURE 9.—SECTION OF THE MCINTOSH & SEYMOUR DOUBLE-ACTING ENGINE.

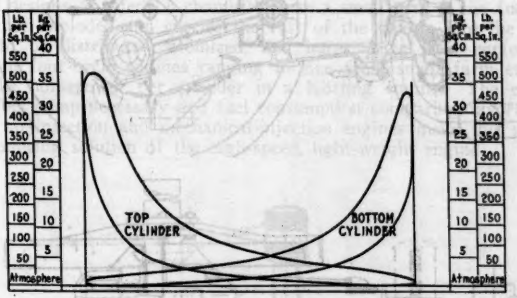



FIGURE 8.—INDICATOR CARDS FOR UPPER AND LOWER SIDES OF CYLINDER.

formance, and even today, after much time and effort have been spent in developing various types of airless injection, a large percentage of the oil-engine builders prefer the assured reliability of air injection.

Space does not permit going into the details of this long chapter of oil engine development, and the reader is referred to the voluminous current literature covering it;[13] suffice it here to say that three different types of airless injection of fuel have been developed that, judging by the fact that they have been adopted by an increasing number of eminent builders of four-cycle engines, bid fair to make the injection-air compressor survive only in the memories of the pioneers.

*Mechanical Injection* employing pump pressures ranging from 2000 to 20,000 pounds per square inch, following the pioneer work of McKechnie of Vickers, Ltd., has been adopted by Hesselman (Polar) in Sweden, M.A.N. and Deutz in Germany, and a number of Pacific Coast oil-engine builders and the Foos Company in this country, as well as by American licensees of the above European firms. The Beardmore Company used it in their high-speed engine[14] (Fig. 10) that made the now famous cross-country run in a Canadian rail car (No. 13, Table 1). Minor modifications embodying the individual preferences of designer in an effort to overcome the originally encountered smoking due to the dripping of fuel and its incomplete combustion, mark the greater number of existing solutions, but all rely essentially upon high pressure on the fuel to bring the fuel charge into sufficiently intimate contact with the body of combustion air to result in complete combustion. The low fuel consumption at all loads and the fine regulation even at no load are the outstanding advantages, in addition to weight reduction resulting from the elimination of air injection and the attached air compressor.

*Injection Chamber.* Another solution of airless injection is the so-called injection chamber (Fig. 11), a type developed out of the original efforts of Brons in Holland to substitute inert products of the combustion chamber for injection air as the carrier and atomizer of the fuel charge. A number of

[13] Among others, a paper by Dr. Lucke in "Mechanical Engineering," Oct., 1921, p. 654; The Diesel Engine of Today (in English), Dr. Naegel, "Zeitschrift des Vereines deutscher Ingenieure," June, 1923; also papers in same journal by G. Eichelberg (Aug. 7, 1926, p. 1079), and by Modersohn (Jan. 5 and 26, 1926, pp. 767 and 891), the latter being abstracted in JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Nov., 1926.

[14] "Oil Engine Power," May, 1926, p. 281.

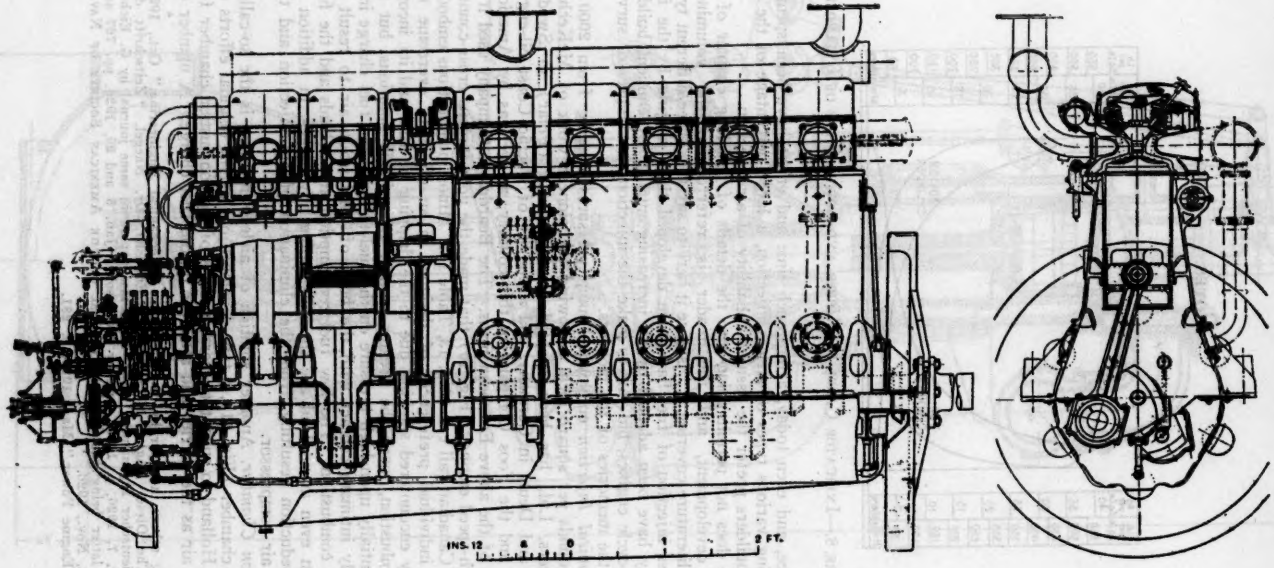



FIGURE 10.—BEARDMORE HIGH-SPEED LOCOMOTIVE ENGINE.

different designs of injection chamber where a small part of the fuel charge is burnt or exploded and carries the rest of the charge into the cylinder proper to be distributed, atomized, and burnt, have been perfected and applied to four-cycle engines ranging in size from small farm engines to 250 brake horsepower per cylinder in a Körting engine. The extremely simple fuel pump necessary and fuel consumption comparing favorably with that of air-injection and mechanical-injection engines make this method a very promising solution of the high-speed, light-weight engine.

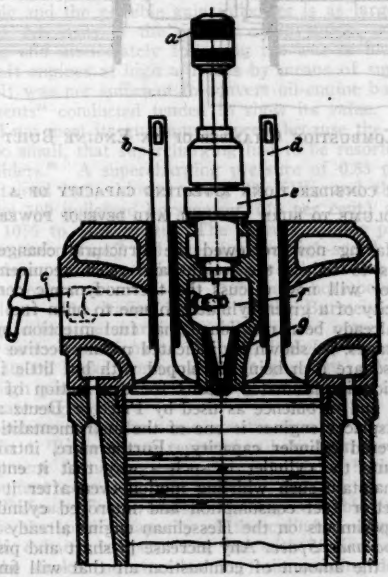



FIGURE 11.—A TYPE OF INJECTION CHAMBER USED BY BENZ CO.

A purely American solution of the air-compressor problem is credited to W. R. Price,[15] who caused the combustion air in the cylinder to distribute the fuel charge over the whole chamber by creating *violent turbulence* within it. Whether the proper atomization of the fuel is really due to impact of the two opposing sprays of fuel or whether the fuel is already sufficiently atomized when it leaves the fuel nozzles, is probably merely of academic interest. The fact remains that low fuel consumption is achieved and a lighter engine results due to the absence of an air compressor. Three American firms build engines of this type (Fig. 12) and a number of locomotives on American railroads use a high-speed engine of the same type as a propelling plant.

[15] *Ibid.*, Nov., 1925, p. 652.

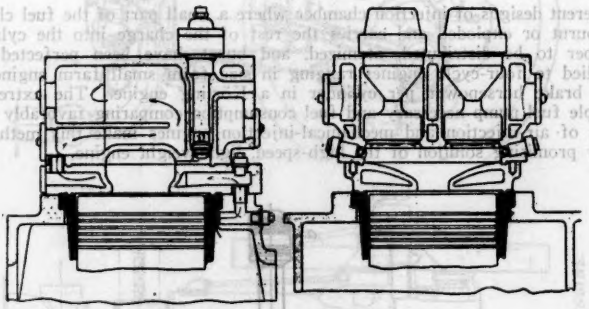


FIGURE 12.—COMBUSTION CHAMBER OF AN ENGINE BUILT BY FALK CO.

THERMODYNAMIC CONSIDERATIONS AFFECTING CAPACITY OF A GIVEN CYLINDER VOLUME TO BURN FUEL OIL AND DEVELOP POWER.

*Turbulence.* Having now reviewed the structural changes employed by various designers to reduce the weight and space requirements of an oil engine, the author will now discuss the thermodynamic considerations affecting the capacity of a given cylinder volume to burn fuel oil and develop power. It has already been mentioned that fuel injection by air has many important advantages, as shown by indicated mean effective pressures up to 150 pounds per square inch being developed with but little if any smoking. When air injection is eliminated means for distribution of the oil charge must be adopted, and turbulence as used by Price or Deutz and Crossley in their so-called displacer engines is one of the instrumentalities employed to improve the lowered cylinder capacity. Furthermore, introduction of the combustion air into the cylinder in such a way that it enters in a spiral whirl tends to maintain such whirling motion even after it is compressed, and results in better fuel consumption and improved cylinder capacity as shown by the experiments on the Hesselman engine already referred to.

*Scavenging Clearance Space.* Any increase in shaft and piston speeds unfavorably affects the amount of combustion air that will find its way into the cylinder since velocities through the valves will also increase, assuming, of course, that the largest valves possible were installed in the first place. In addition the cylinder interior becoming hotter, the temperature of the incoming air increases and the amount (by weight) decreases still farther. Furthermore, normally not less than 7 per cent of the cylinder volume is filled with hot products of combustion from the preceding cycle that cannot be displaced by the upward-moving piston. There is also usually a slight overlap between the closing of the exhaust valve and the opening of the inlet valve which permits a certain amount of exhaust gases to get into the cylinder since the exhaust header is under slight pressure. Thus the loss of cylinder capacity is quite apparent. The recovery of this loss was the subject of experiments both in this country by the Falk Company[16] and in Germany. The inlet valves for this purpose are connected to a closed manifold, where a pressure of about 15 pounds absolute is maintained by a blower. This will result in a thoroughly scavenged clearance volume due

[16] "Motorship" (N. Y.), Oct., 1924, p. 784.

to the above-mentioned overlap in valve timing and the fact that all of the cylinder volume is filled with pure air of atmospheric or slightly higher pressure. At the last meeting of the V.D.I. (Society of German Engineers) the statement was made that such scavenging of clearance volume permits of a 38 per cent increase in power as compared with unscavenged operation.[17] Such remedy will involve of course the use of some blower, whether of the rotary or Roots[18] or centrifugal[19] type, which may be either geared as the first one or driven by an electric motor, steam turbine, or exhaust-gas turbine.[20]

This apparent additional complication will hardly be objectionable if the blower is reliable and the possible gain in power is as large as claimed.

*Supercharging.* Necessity is the mother of invention, and though much was done during and immediately following the war to increase the power output of aircraft engines at high altitudes by means of supercharging, this stimulus of itself was not sufficient to convert oil-engine builders to its use, though experiments[21] conducted tended to show its value. It was when a motorship could not meet its guaranteed speed, because the engines installed proved to be too small, that supercharging had to be resorted to to save the face of the builders.[22] A supercharging pressure of 0.85 pound per square inch was maintained by a blower using 33 KW., and this increased the engine output by 500 indicated horsepower (16 per cent) and the speed of the ship from 10½ to 11½ knots. The success of this practice led to its adoption for the sister ships[23] and later for the high-speed gear-driven plant for ship propulsion to reduce the specific weight of the installation[24] by increasing engine output from 1350 to 1750 brake horsepower (30 per cent) —No. 14, Table 1. The Italian licensee of the Burmeister & Wain Company resorted also to supercharging to increase the output of double-acting engines of the same dimensions as in the *Gripsholm*, but using 8 cylinders, from 9000 to 10,000 brake horsepower (11 per cent)—No. 15, Table 1—retaining the high piston speed and driving the injection-air compressor by means of separate engines.[25]

Engines of substantially the same construction as used in the German submarines—No. 1, Table 1—are reported to have been installed with supercharging equipment furnishing air of substantially higher pressure, and having the resultant extremely low specific weight.[26]

*Increase in Exhaust Temperatures.* The objectionable feature of any effort to increase the specific capacity of the cylinder volume is the undesirable increase in exhaust temperatures, but with efficient combustion at the beginning of the stroke (no after-burning) the existing designs of cylinder heads and exhaust valves discussed earlier are apparently capable of coping with the situation arising from supercharging with blower pressures up to 15–16 pounds per square inch. Further increase in supercharging pressure if considered from the standpoint of fuel economy is very limited, since the energy of the exhaust gases would soon be insufficient to drive the super-

[17] Prof. Magg in "Zeitschrift des Vereines deutscher Ingenieure," Aug. 7, 1926, p. 1090.
[18] See N. A. C. A. Report No. 230.
[19] See Dr. Moss's paper in "Mechanical Engineering," Mid-Nov., 1925, p. 1078.
[20] "Motorship" (N. Y.), March, 1925, p. 203; "The Motorship" (British), May, 1926, p. 61.
[21] By Dr. Riehm of M.A.N.; see "Zeitschrift des Vereines deutscher Ingenieure," Aug. 4, 1923, p. 763, abstracted in "The Motorship" (British), Nov., 1924, p. 294.
[22] "Motorship" (N. Y.), Mar., 1921, p. 179.
[23] "The Motorship" (British), Nov., 1924, p. 291 and 294.
[24] *Monte Sarmiento* and *Monte Olivia*. "Motorship" (N. Y.), Dec., 1924, p. 308, and Feb., 1925, p. 100.
[25] *Saturnia* and *Urania*. "Motorship" (N. Y.), Feb., 1926, p. 98.
[26] "The Motorship" (British), Aug., 1926, p. 181.

charging blower, and fuel consumption would suffer. But for the same reason the steam engine with atmospheric exhaust should have no place in our industrial life, yet countless thousands of horsepower are developed in such manner in locomotives alone. Is it not logical, then, to conclude, that if substantial increase in the horsepower of an oil engine without endangering its reliable performance is possible by supercharging, supercharging is the proper solution for the propelling plant of a warship, for instance, where the maximum speed is needed but rarely and where every pound of weight counts? The details of any such plant may include variable clearance volume with auxiliary fuel injection, or two different clearance volumes for the two ends of a double-acting cylinder, with one side permitting a high degree of supercharging or running idle when it can be spared, while the other works on the standard self-induction cycle. Success in such a research would be a most generous reward for the patience and foresight of an engineering executive and the perseverance of his staff.

## II—TWO-CYCLE ENGINES.

As has been mentioned in the opening paragraphs of this paper, oil engines using the two-stroke cycle burn roughly twice the amount of fuel for the same cylinder volume and develop about twice the power. Hence the working cylinder weight per horsepower is considerably reduced, provided there are no practical obstacles to using such a method to improve the efficiency of material.

There have been difficulties, and many of them, so many in fact that a large number of two-cycle-engine manufacturers, especially of engines of moderate output, have been and are now content with much greater specific engine weight, as evidenced by the small brake mean effective pressure of their ratings, as long as the fundamental simplicity of two-cycle design and reliability of engine performance are maintained. The chief difficulty lies in accomplishing that which takes the four-cycle engine two out of every four strokes, or one-half of its time, to do, namely, scavenging and charging. The real problem is to do this in as short a time as possible and as efficiently—and even possibly more efficiently—as it is done in the four-cycle engine. In the absence of such conditions as are present in the four-stroke cycle, a great many widely different solutions have been proposed; and a considerable number of them have survived and are being further developed to meet the more acute competition of the four-cycle engine and the more severe requirements of the time as to weight and space.

### METHODS OF SCAVENGING.

Methods of scavenging may be divided into two groups, namely, those where scavenging air charges the cylinder without changing its direction of flow, driving the exhaust gases ahead of it (like in a uniflow steam engine), and those where it reverses or otherwise changes its direction of flow. The first may be further subdivided into those employing valves in the cylinder head and ports at the bottom of the cylinder, and those using ports at both ends of the cylinder. The second group is confined only to the use of ports at the lower end of the cylinder, though a number of modifications of the relative positions of the two types of ports are in existence.

*Valves in Head.* A number of high-speed engines for U. S. submarines[37] of the M.A.N. step-piston type using one and later two valves in the head for scavenging have demonstrated that as far as cylinder scavenging is con-

[37] JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Aug., 1925, p. 583.

cerned this type operated quite reliably. The use of a stepped piston for pumping scavenging air gave rise to difficulties that resulted in the adoption by the M.A.N. company of the further developed, hence more reliable, four-stroke cycle for the German submarine engines, then badly needed due to the breaking out of the war. Valve scavenging when applied to cylinders of large diameter such as those of the M.A.N. type on the U. S. naval tanker *Maumee*[28] (No. 1, Table 2), was not successful due to cylinder heads failing under the excessive heat stresses. Changes in design and material did not solve the *Maumee's* troubles, but some other builders when confronted with similar experience were more successful in finding remedies.

The largest cylinder output so far built and incorporated into a full-size engine—2000 brake horsepower—is that of the M.A.N. double-acting 12,000-horsepower engine (No. 2, Table 2) for warship propulsion;[29] this engine used valves for scavenging, although they were located on the periphery of the cylinder and not in the cylinder heads to save headroom. The largest stationary engine built in this country (for the Panama Canal)[30]—No. 3, Table 2—is of this type, as well as many other large engines of the same type, for example, the Nordberg-Carels. Here the solution appears to have been found in the use of a two-piece cylinder head (Fig. 13). The inner

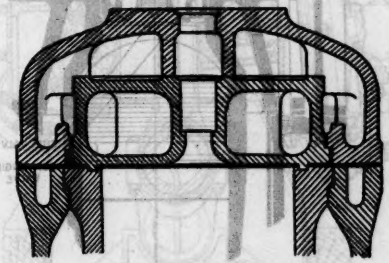


FIGURE 13.—TYPE OF HEAD USED ON NORDBERG-CARELS ENGINE.

one, exposed to the heat of the combustion, is made of cast steel, as simple as possible and free to expand, whereas the cylinder-head bolts pass through the heavy outer head made of cast iron and serving as part of scavenging receiver. Another successful solution is that by the Krupps, who introduced a separate water-cooled shield to protect the heat-exposed surface of the cylinder head, a device they have retained even in their double-acting port-scavenging engine (Fig. 26), where the need of it, it would seem, is no longer apparent, and where also the two-piece cylinder head is used. This principle of providing free expansion for the parts of cylinder heads and cylinders is recognized in most of the new designs of two-cycle engines, and its adoption is probably largely responsible for their success.

An American engineer, Arthur West, is responsible for the Bethlehem Steel Company's solution of the scavenging problem[31] (No. 4, Table 2). A single centrally located scavenging valve of very simple dished shape

[28] JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Nov., 1915, p. 794.
[29] See "Zeitschrift des Vereines deutscher Ingenieure," Dec. 8 and 22, 1923; "Motorship" (N. Y.), Jan., Feb., and Mar., 1924.
[30] "Oil Engine Power," Feb., 1925, p. 77.
[31] "Motorship" (N. Y.), March, 1924, p. 198; also March, 1926, p. 211.

admits the scavenging air (Fig. 14). A number of such engines are said to have been built both for marine and stationary power plants, and their successful operation will no doubt do much toward eradicating any existing prejudice against such an original and daring solution of the scavenging problem for the large oil engine.

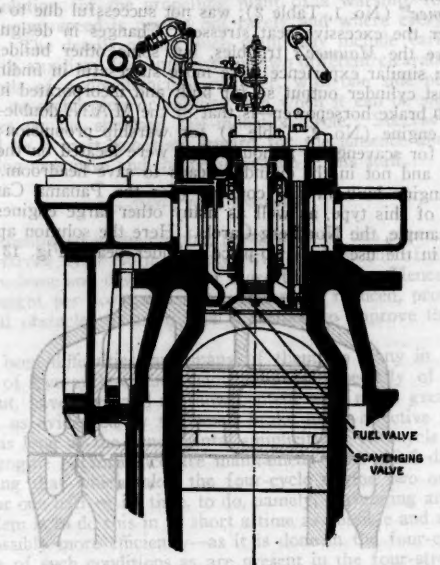



FIGURE 14.—SECTION THROUGH CYLINDER AND HEAD OF BETHLEHEM ENGINE.

Valve-in-the-head scavenging applied to cylinders of small diameter is not accompanied by any serious heat troubles, as was shown by the high-speed submarine engines. Its application to building an oil engine suitable for aircraft (Fig. 15), as was done by Attendu,[28] will, it is hoped, contribute much to the solution of high-speed light-weight oil-engine problems (No. 5, Table 2).

*Opposed-Piston Engines.* Ports have also been used to obtain the same individual flow of scavenging air and with a much smaller loss due to throttling of the air passing through the restricted areas of the admission organ. The size of valves is limited by the space available in the cylinder head and the lift by the acceleration of the valve mass which it is safe to use, whereas ports may occupy all of the periphery of the cylinder, and their height need be only somewhat smaller than for the exhaust ports. Furthermore, ports offer the opportunity of so shaping them that initial whirling motion by the scavenging air is obtained. This not only tends to better displacement of the exhaust gases but also creates turbulence which is

[28] "Journal S.A.E.," Feb., 1926, p. 1.

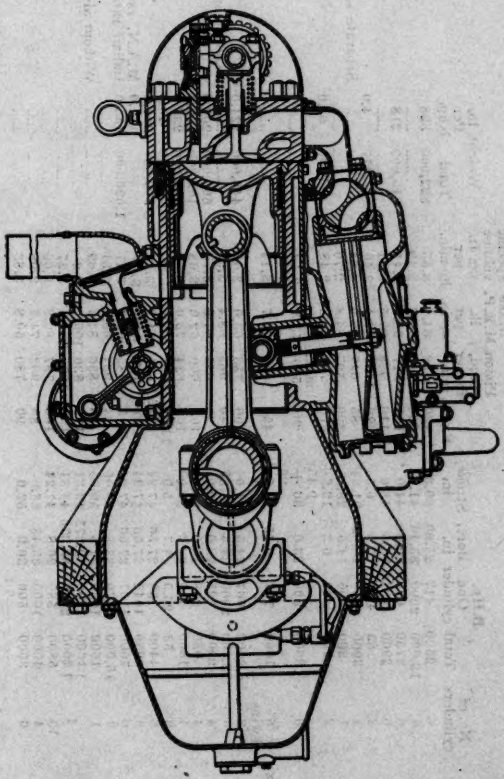
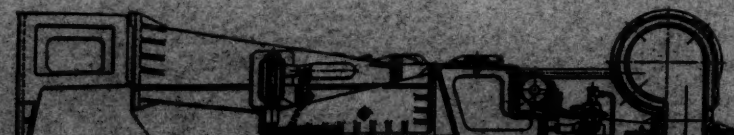


FIGURE 15.—SECTION THROUGH ATTENDU AIRCRAFT ENGINE.

TABLE 2 DATA ON TWO-CYCLE ENGINES[1]

| No. | Type of engine | S.A. or D.A. | No. of cylinders | B.Hp. Total | B.Hp. One cylinder | Bore, in. | Stroke, in. | R.p.m. | Piston speed, ft. per min. | Brake M.E.P. lb. per sq. in. | Specific volume cu. ft. per hp-min. | Weight, lb. Total | Weight, lb. Per b.hp. | Remarks |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | M.A.N.-*Maumee* | S.A. | 6 | 2500 | 417 | 25.20 | 39.37 | 130 | 853 | 64.8 | 3.55 | 822,000 | 328 | |
| 2 | M.A.N. (1917) | D.A. | 6 | 12,000 | 2000 | 33.46 | 41.34 | 160 | 1100 | 71.7 | 3.19 | ..... | ..... | |
| 3 | Nordberg-Carels | S.A. | 6 | 3750 | 625 | 29.0 | 44.0 | 135 | 917 | 68.5 | 3.41 | 815,000 | 218 | |
| 4 | Bethlehem | S.A. | 6 | 2900 | 483 | 25.5 | 48.0 | 116 | 928 | 67.5 | 3.41 | ..... | ..... | |
| 5 | Attendu | S.A. | 2 | 85 | 42.5 | 5.5 | 6.5 | 1630 | 1755 | 67.4 | 3.40 | 477 | 4.9 | |
| 6 | Sun-Doxford | O.P. | 4 | 3000 | 750 | 23.62 | 91.34[2] | 85 | 647 | 87.0 | 2.64 | ..... | ..... | Separate scavenging blower |
| 7 | Doxford-Submarine | O.P. | 1 | 300 | 300 | 14.5 | 28.3[2] | 300 | 707 | 84.8 | 2.70 | ..... | ..... | |
| 8 | Junkers (1923) | O.P. | 2 | 120 | 60 | 6.3 | 12.6+ 9.45 | 375 | 788 max. | 92.3 | 2.48 | 7700 | 64.0 | |
| 9 | Palmer-Fullagar | O.P. | 6 | 3000 | 500 | 23.0 | 36 + 36 | 90 | 540 | 73.5 | 3.12 | ..... | ..... | |
| 10 | Knudsen-Baldwin | Duplex Twins | 6 | 1000 | 167 | 9.75 | 13.5 + 13.5 | 450 | 1010 | 72.7 | 3.14 | ..... | ..... | |
| 11 | North-British | D.A. | 3 | 2000 | 667 | 24.5 | 44.0 | 100 | 733 | 63.0 | 3.6 | 448,000 | 224.0 | |
| 12 | Krupp-1917 | D.A. | 6 | 12,000 | 2000 | 34.4 | 41.35 | 140 | 965 | 68.4 | 3.35 | ..... | ..... | 10,600 b.hp. only obtained on test |
| 13 | Worthington | D.A. | 4 | 2900 | 725 | 28.0 | 40.0 | 95 | 633 | 64.8 | 3.74 | 600,000 | 207.0 | |
| 14 | Worthington | D.A. | 4 | 950 | 238 | 16.0 | 24.0 | 165 | 660 | 62.0 | 3.68 | 162,000 | 170.0 | |
| 15 | Nobel | S.A. | 4 | 1600 | 400 | 26.6 | 36.2 | 105 | 634 | 75.0 | 3.06 | 380,000 | 238.0 | |
| 16 | Peugeot-Tartrais | S.A. | 2 | 53 | 26.5 | 4.7 | 5.9 | 1450 | 1430 | 70.5 | 3.25 | 530 | 10.0 | |
| 17 | M.A.N. | D.A. | 6 | 4400 | 733 | 27.56 | 47.24 | 84 | 661 | 64.8 | 3.53 | 900,000 | 204.0 | |
| | | | 6 | 6250 | 1042 | 27.56 | 47.24 | 120 | 945 | 64.6 | 3.55 | | 174.0 | M.A.N. rating |
| 18 | M.A.N. | D.A. | 6 | 7000 | 1167 | 27.56 | 47.24 | 125 | 985 | 69.2 | 3.31 | 1,090,000 | 155.0 | Italian rating (M. S. *Augustus*) |
| 19 | Blohm & Voss-M.A.N. | D.A. | 9 | 15,000 | 1670 | 33.8 | 59.05 | 94 | 934 | 69.2 | 3.31 | ..... | ..... | |
| 20 | Krupp | D.A. | 1 | 1600 | 1600 | 31.5 | 59.05 | 85 | 836 | 84.4 | 2.69 | ..... | ..... | Without air compressor |
| 21 | Sulzer Bros. (1914) | S.A. | 1 | 1600 | 1600 | 39.375 | 43.31 | 115 | 830 | 105.0 | 2.9 | ..... | ..... | |
| | | | | 2000 | 2000 | 39.375 | 43.31 | 150 | 1070 | 101.5 | 2.27 | ..... | ..... | |
| 22 | Sulzer Bros. | S.A. | 10 | 5800 | 580 | 26.8 | 47.24 | 100 | 787 | 86.2 | 2.66 | ..... | ..... | |
| 23 | Brown-Sulzer | S.A. | 4 | 4000 | 1000 | 35.43 | 63.0 | 86 | 903 | 74.2 | 3.08 | ..... | ..... | |
| 24 | Busch-Sulzer | S.A. | 6 | 3000 | 500 | 30.0 | 52.0 | 90 | 780 | 59.8 | 3.83 | ..... | ..... | |

[1] Ten per cent allowance made for piston-rod area.
[2] Combined stroke.



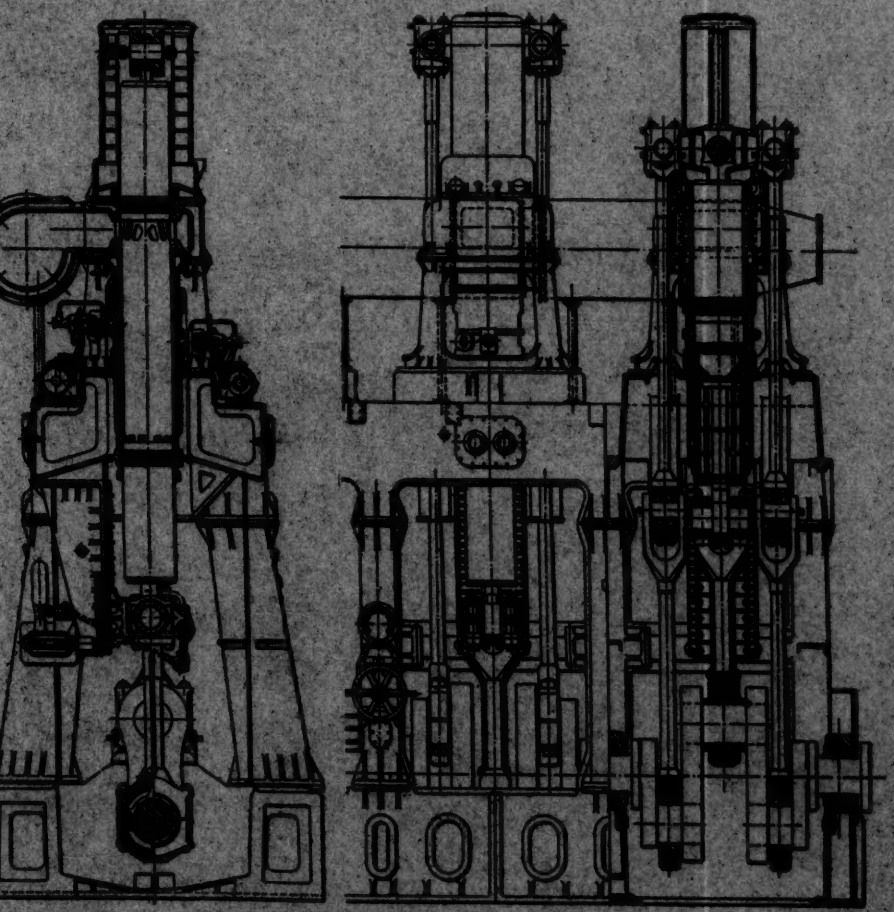


FIGURE 16.—2000-B.H.P. SUPE-DOXFORD ENGINE.

FIG. 20.—40 H.P. Sun-Dowson Engine.

so important for complete combustion, especially when no injection air is used. Opposed pistons inherited from gas-engine practice have been used by Junkers and Fullagar, each of the pistons controlling one set of ports. The sliding cylinder, the stroke of which is only sufficient to uncover the upper or scavenging ports, has been incorporated by Maclagan in the design of North British type of oil engine.

*The Junkers Engine* possesses other features of exceptional merit such as complete balance of reciprocating masses, relief of cylinder and engine framing from explosive stresses, and absence of cylinder heads, which also tend to reduce its specific weight as well as does the employment of the two-stroke cycle. However, the German licensees of Junkers failed to get results chiefly because of their inexperience in oil-engine building in general, and only Doxfords in England under the guidance of Junkers' associate have achieved commercial success. The Doxford engine,[33] built in this country by Sun Shipbuilding & Dry Dock Company (Fig. 16), has been further lightened by the elimination of the injection-air compressor and the substitution of mechanical injection (No. 6, Table 2). During the war an effort was made to adopt this type of engine for submarine propulsion and an experimental cylinder was built and tested[34] in the Admiralty Engineering Laboratory without, however, any further results due to cessation of hostilities (No. 7, Table 2). This foreign success led Junkers to work out a new design[35] for his engine (Fig. 17), that also employed mechanical injection and in compactness of arrangement suggested a low-pressure steam engine (No. 8, Table 2). The lowest fuel economy ever achieved in any oil engine, 0.345 pound per brake horsepower-hour, is claimed to have been obtained with this engine. A serious objection to this type of engine is the fact that it does not lend itself to very high rotative speeds due to the inevitable heavy weight of the upper piston and masses reciprocating with it unless two separate crankshafts are employed, direct-connected to the load, such as propellers in the aircraft type of engine, said to have been developed by Junkers.

A modification of the Junkers engine aiming at doing away with the multiplicity of cranks is the Fullagar engine, built by the Camellaird-Fullagar Co. and its licensees[36] (No. 9, Table 2). Here the cylinders are built in pairs close together with one crank for each, and each crank receives alternately the pressures of one upper and one lower piston, the two being connected with each other by diagonally slanted rods. Air injection of fuel has been retained by the builders of this type of engine. Excellent fuel consumption (0.39 pound per brake horsepower-hour) and very economic utilization of the cylinder volume, with a corresponding light weight and compactness, have been obtained in this type of engine, and a very large engine—10,000 brake horsepower in 8 cylinders[37]—has been designed, but so far not built. However, the reported removal of engines of this type from the electric-driven motorship *La Playa*[38] without any cause being specifically published, leads to the suspicion that the existing design is not free from weak spots imperiling the reliable performance of the engine.

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[33] "Motorship" (N. Y.), Sept., 1924, p. 652.

[34] "The Engineer," May 4, 1923, p. 489.

[35] "Zeitschrift des Vereines deutscher Ingenieure," Oct. 31, 1925, p. 1369; also "Oil Engine Power," Dec., 1925, p. 696.

[36] "The Motorship" (British), March, 1924, p. 470; Sept., 1924, p. 200; Oct., 1924, p. 231; and "Motorship" (N. Y.), Dec., 1923, p. 854, and Feb., 1924, p. 118.

[37] "The Motorship" (British), Nov., 1924, p. 276.

[38] "The Motorship (British), Nov., 1923, p. 281; and "Motorship" (N. Y.), Aug., 1926, p. 597.

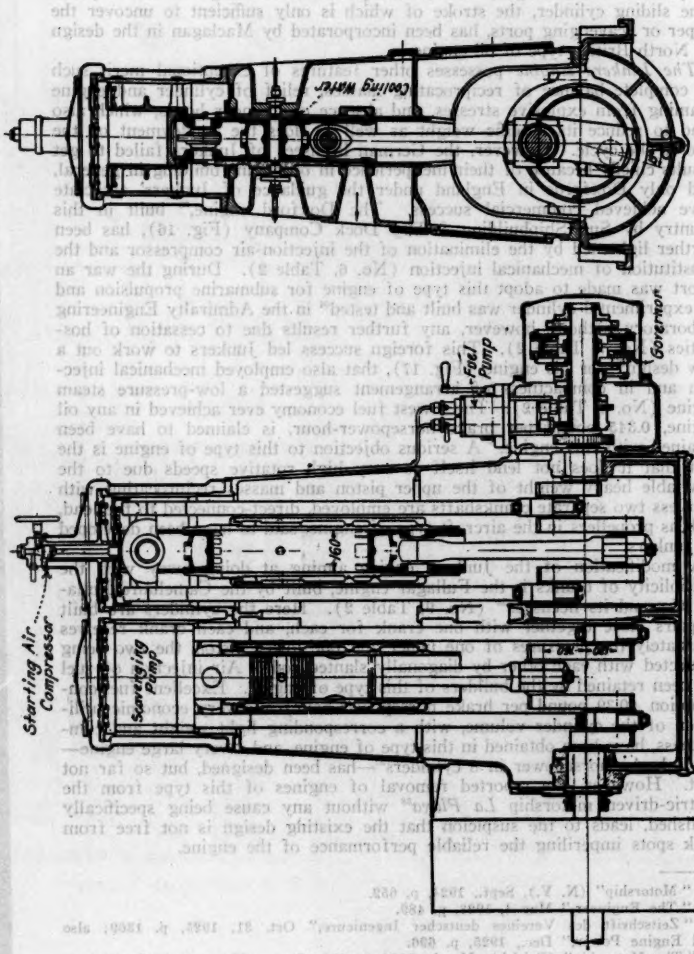


FIGURE 17.—NEW DESIGN (1923) OF JUNKERS ENGINE

A further modification of the Junkers engine is the Knudsen engine (Fig. 18) installed by the Baldwin Locomotive Co. in their locomotive[39] (No. 10, Table 2). However, here most of the above-named advantages of the opposed-piston designs have been sacrificed and only the one-direction (uniflow) scavenging retained, although the disadvantage of unequal reciprocating weights also disappears and the engine has been built as a high-speed engine. Furthermore, by placing the two crankshafts so close together their interconnection by means of gears has become feasible and has been employed. Whether the cylinder head containing the common combustion chamber will stand up under severe heat stresses, whether air cooling of

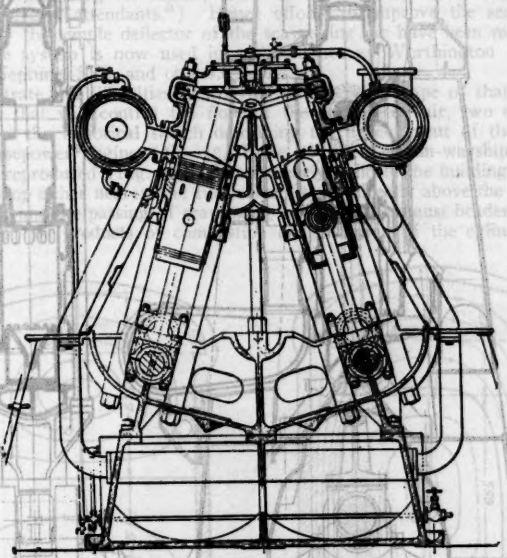

FIGURE 18.—CROSS-SECTION OF KNUDSEN OIL ENGINE.

pistons is sufficient, and whether torsional-vibration problems arising in such a high-speed gear-driven machine, as mentioned already, have been properly taken care of will be answered by extended operation of this engine unit in service.

*Sliding-Cylinder Engine.* To obtain the advantages of uniflow scavenging with scavenging ports of large area, Maclagan proposed the sliding-cylinder type of engine, and a double-acting engine of this type consisting in reality of two separate single-acting cylinders in tandem has been built by the North British Diesel Engine Works[40] (Fig. 19), and two more have been installed in motorships (No. 11, Table 2). Whether such radical departure from more familiar forms is justified by the results obtained or that can be

[39] "Oil Engine Power," Sept., 1925, p. 523.
[40] "The Motorship" (British), May, 1923, p. 42, and Nov., 1924, p. 284.

FIGURE 17.—NEW DESIGN (1923) OF JUNKERS ENGINE

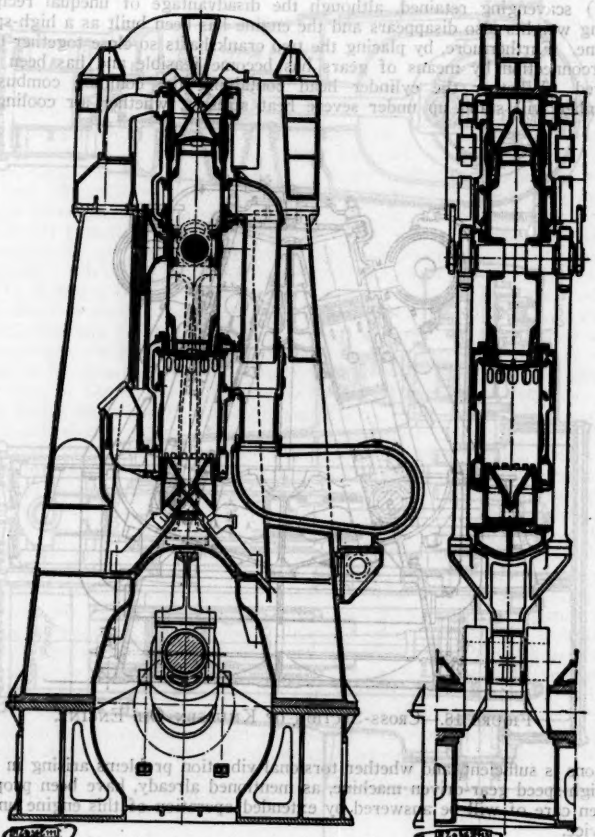

FIGURE 19.—SECTIONS OF NORTH BRITISH DOUBLE-ACTING ENGINE.

obtained with this construction, time alone can tell. It does appear, however, that the severe handicap of the natural prejudice of the operating engineer to radical departures will long hamper its development.

*Port Scavenging.* The possibility of dispensing with all valves is a most valuable feature of a two-cycle engine, long appreciated by builders of small engines for motor boats. Simplicity of such arrangement while valuable in rather small oil engines where it is often paid for by substantial rise in fuel consumption and higher specific weight, becomes particularly important in large engines, where the valves and valve gear and its drive become important problems of design and maintenance. (A sticking scavenging valve was responsible for the explosion in the scavenging receiver of the large Nuremberg oil engine, resulting in the ruin of the engine and death of several attendants.[41]) Hence efforts to improve the scavenging obtained by the simple deflector of the scavenging air have been made, and this simple system is now used in such engines as Worthington (double-acting), Neptune, Still, and others.

To illustrate the difficulties in arriving at the right shape of that part of the piston that will control the flow of the scavenging air, two diagrams taken from the historical sketch describing the development of the Krupp 12,000-horsepower engine (No. 12, Table 2) for German-warship propulsion[42] are reproduced (Fig. 20). These clearly indicate the building up of a whirling loop either nearest the fuel valve (*a*) or directly above the receding piston (*b*), and the passing of scavenging air into the exhaust header without driving out the products of combustion and charging of the cylinder with

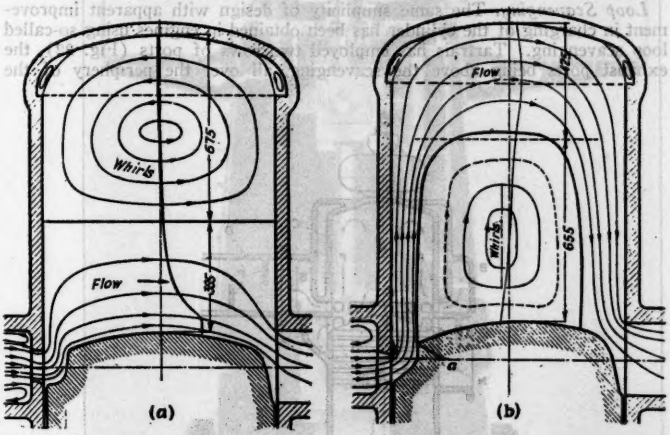


FIGURE 20.—STREAMS OF SCAVENGING AIR AND WHIRLING LOOP OF EXHAUST GASES IN THE KRUPP 12,000-H.P. ENGINE (*a*) WITH ORIGINAL PISTON AND (*b*) AS MODIFIED.

[41] The Diesel Engine of Today, by Dr. Naegel, "Zeitschrift des Vereines deutscher Ingenieure, June, 1923 (in English), also same journal Dec. 8 and 22, 1923, and "Motorship" (N. Y.), Jan., Feb., and Mar., 1924.

[42] "Zeitschrift des Vereines deutscher Ingenieure, Nov. 15, 1924, p. 1203.

fresh combustion air. The inevitable result of such action was that the engine could not develop its designed horsepower due to lack of air in the cylinder and smoky exhaust with high exhaust temperatures. Experiments with the Scott-Still engine scavenging have also shown that.[43]

The new Worthington double-acting engine[44] (Fig. 21) has enjoyed much deserved publicity both here and abroad mainly for its extreme simplicity, due largely to its method of scavenging. Its rating at first glance appears conservative (Nos. 13 and 14, Table 2), both as to piston speed and mean effective pressure, but the rather high fuel consumption obtained on test of the Shipping Board engines tends to indicate that the low efficiency of the method of scavenging and charging of the cylinder would hardly permit a much higher rating. But even with this handicap its specific weight is materially lower than that of any of its competitors on the S.B. program, and for the smaller engine recently installed[45] it is as low as in any engine of its class.

The same type of scavenging, with the addition of light automatic valves to prevent the cylinder contents from entering the scavenging receiver while exhaust by excess pressure in the cylinder takes place, is used in Nobel and Fiat engines. This feature permits the scavenging ports to be made of equal height with the exhaust ports, resulting in much better charging of the cylinder and consequently a lighter engine (No. 15, Table 2). An experimental high-speed, two-cylinder, 1000-horsepower engine employing this type of scavenging has been built and successfully tested by the Fiat Co. for the purpose of developing a high-power engine for the Italian submarine,[46] but unfortunately no further data about it are available.

*Loop Scavenging.* The same simplicity of design with apparent improvement in charging of the cylinder has been obtained in engines using so-called loop scavenging. Tartrais has employed two rows of ports (Fig. 22), the exhaust ports being above the scavenging, all over the periphery of the

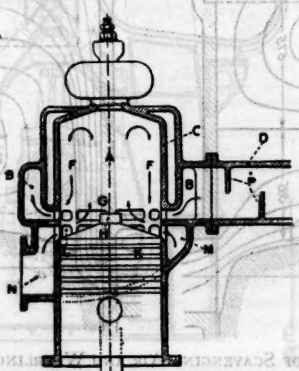


FIGURE 22.—SCAVENGING METHOD USED BY TARTRAIS IN PEUGEOT ENGINE.

[43] "Motorship (N. Y.), Dec., 1923, p. 871.
[44] "Motorship" (N. Y.), April, 1926, p. 270.
[45] "Marine Engineering," Oct., 1926, p. 577.
[46] "The Motorship" (British), Oct., 1926, p. 247.

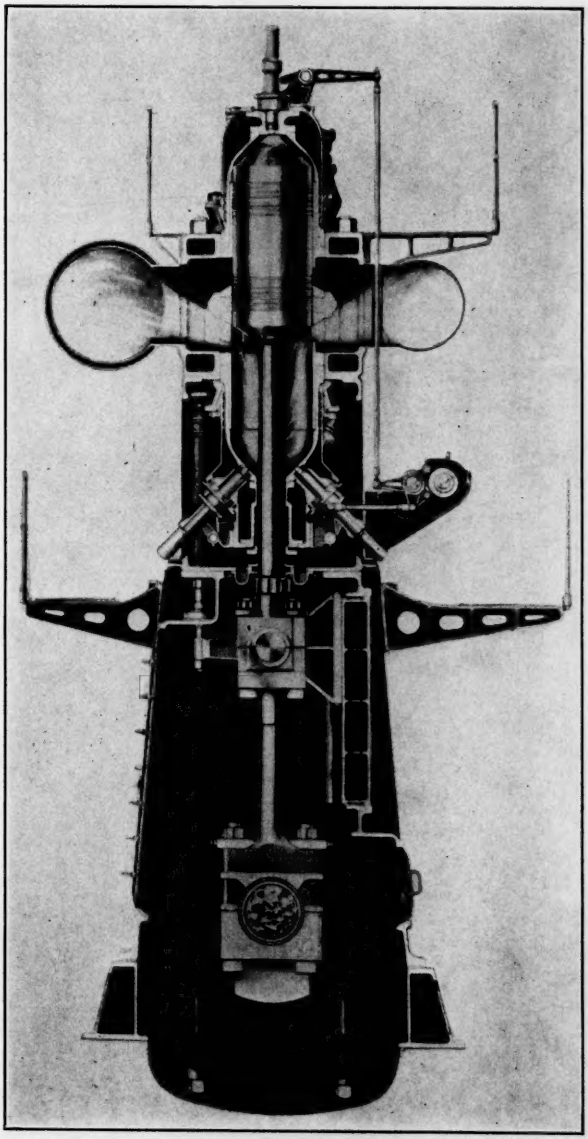

FIGURE 21.—CROSS-SECTION OF THE WORTHINGTON DOUBLE-ACTING ENGINE.

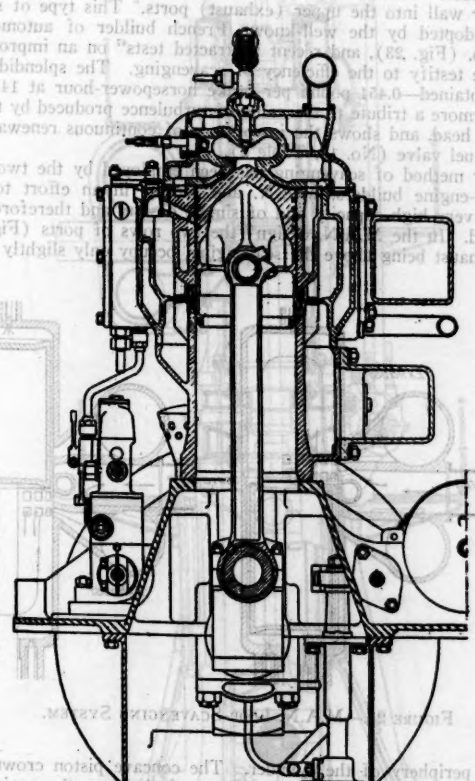

FIGURE 23.—SECTION THROUGH CYLINDER OF PEUGEOT ENGINE.

cylinder. By giving a suitable shape to the piston crown he makes the opposing ports produce two parallel streams of scavenging air in the center of the cylinder. These are deflected by the cylinder head and return along the cylinder wall into the upper (exhaust) ports. This type of scavenging has been adopted by the well-known French builder of automobiles, the Peugeot Co. (Fig. 23), and recent protracted tests[47] on an improved model of this type testify to the efficiency of scavenging. The splendid fuel consumption obtained—0.451 pound per brake horsepower-hour at 1450 R.P.M. —is furthermore a tribute to the value of turbulence produced by this design of cylinder head, and shows the importance of continuous renewal of fresh air at the fuel valve (No. 16, Table 2).

A similar method of scavenging has been employed by the two foremost German oil-engine builders, M.A.N. and Krupp, in an effort to create a cylinder of very high capacity and of simple design, and therefore inexpensive to build. In the M.A.N. design[48] the two rows of ports (Figs. 24 and 25), the exhaust being above the scavenging, occupy only slightly over one-

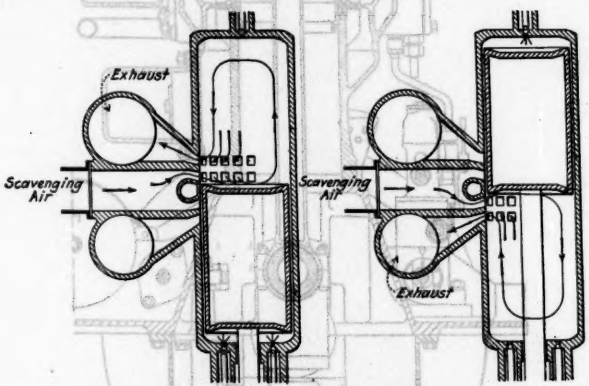


FIGURE 24.—M.A.N. LOOP-SCAVENGING SYSTEM.

half of the periphery of the cylinder. The concave piston crown and the shape of scavenging ports produce a column of air along the unpierced part of the cylinder wall, which is deflected by the cylinder head and returns along the opposite wall to the exhaust ports. Krupp's modification[49] (Fig. 26) consists of having the higher exhaust ports along the axis of the engine, whereas the scavenging ports are on one side only, so that the scavenging-air column is also guided by the opposite wall but is split in two columns directed toward each set of exhaust ports. The scavenging efficiency is evident from the figures in the table, especially when allowance is made for the natural conservatism displayed by the rating of the first engines of the type.

[47] "Mechanical Engineering," June, 1926, p. 615.
[48] "Motorship" (N. Y.), May, 1926, p. 367 and Oct., 1925, p. 747.
[49] "Werft-Reederei-Hafen," Aug. 7, 1926, p. 388.

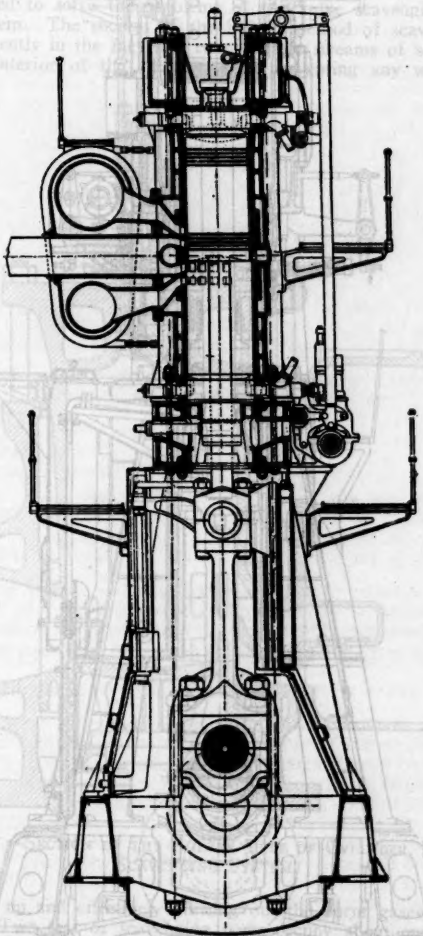

FIGURE 25.—SECTION OF M.A.N. DOUBLE-ACTING ENGINE.

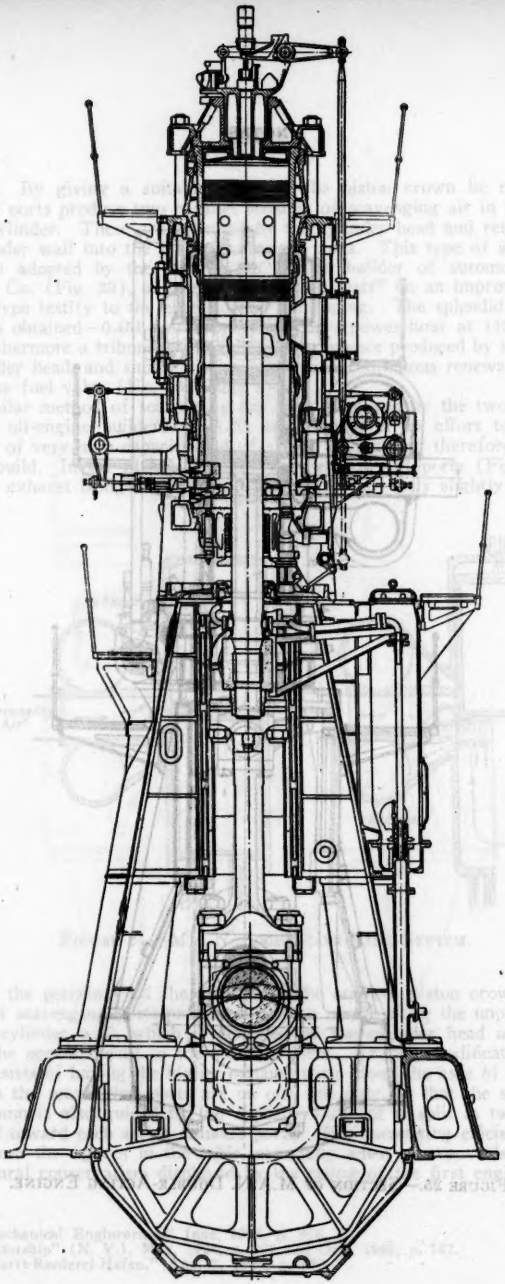

FIGURE 26.—SECTION OF KRUPP DOUBLE-ACTING ENGINE.

*Sulzer Type of Scavenging.* A distinct type of scavenging was developed by Sulzer Bros. when this firm, realizing the possibilities of the two-stroke cycle, attempted to solve the problems of the valve scavenging originally adopted by them. The success of the Sulzer method of scavenging (Fig. 27) lies apparently in the fact that two separate streams of scavenging air traverse the interior of the cylinder, thus preventing any whirling loops

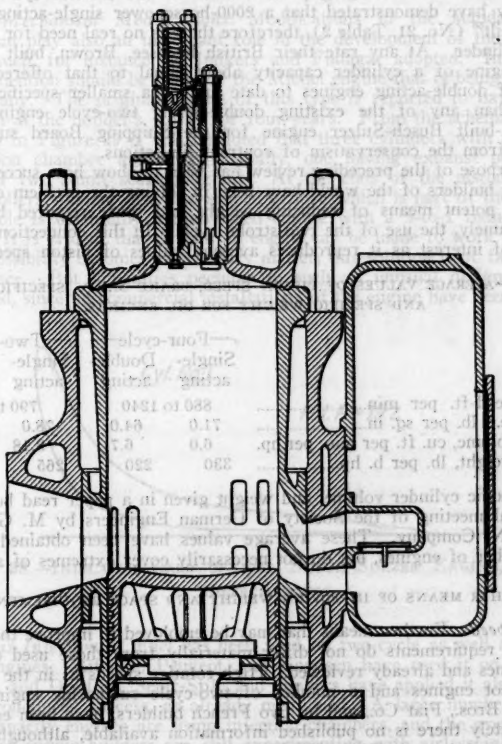

FIGURE 27.—SECTION OF THE SULZER TYPE OF CYLINDER, SHOWING SCAVENGING SYSTEM.

from building up and effectively blowing out the burnt gases into the exhaust ports. Two sets of scavenging ports occupy about one-half of the cylinder periphery and have suitable guiding surfaces so as to scavenge both the upper part of the cylinder immediately at the fuel valve and the space above the piston. The exhaust ports occupy the opposite side of the cylinder. The upper scavenging ports are opened by the piston before the exhaust and are therefore equipped with light automatic valves (in new designs), whereas the lower are beyond the danger of admitting exhaust

gases into the scavenging receiver and are piston-controlled only. An additional advantage of this type of scavenging is the ability to improve charging of the cylinder by filling it with fresh air from the scavenging receiver after the exhaust ports are closed, up to the pressure in the receiver. This supercharging feature will be dealt with later.

Sulzer Bros. appear to be consistently maintaining the position that the efficiency of their scavenging permits them to use smaller cylinders and, since they have demonstrated that a 2000-horsepower single-acting cylinder can be built[80] (No. 21, Table 2), therefore there is no real need for a double-acting cylinder. At any rate their British licensee, Brown, built a single-acting engine of a cylinder capacity about equal to that offered by any builder of double-acting engines to date and of a smaller specific cylinder volume than any of the existing double-acting two-cycle engines. The American-built Busch-Sulzer engine for the Shipping Board suffers apparently from the conservatism of contract limitations.

The purpose of the preceding review has been to show how successful the oil-engine builders of the world have been in solving the problem of one of the most potent means of reducing weight and space required by an oil engine, namely, the use of the two-stroke cycle. In this connection Table 3 may be of interest as it reproduces average values of piston speed, brake

TABLE 3.—AVERAGE VALUES OF PISTON SPEED, BRAKE M.E.P., SPECIFIC VOLUME AND SPECIFIC WEIGHT FOR OIL ENGINES.

| | Four-cycle | | Two-cycle | |
|---|---|---|---|---|
| | Single-acting | Double-acting | Single-acting | Double-acting |
| Piston speed-ft. per min. | 880 to 1240 | | 790 to 950 | |
| Brake m.e.p. lb. per sq. in. | 71.0 | 64.0 | 78.0 | 62.5 |
| Specific volume, cu. ft. per min. per hp. | 6.0 | 6.7 | 3.18 | 3.53 |
| Specific weight, lb. per b. hp. | 330 | 220 | 265 | 200 |

m.e.p., specific cylinder volume, and weight given in a paper read before the last annual meeting of the Society of German Engineers by M. Gercke of the M.A.N. Company. These average values have been obtained from a large number of engines, but do not necessarily cover extremes of any type.

## FURTHER MEANS OF IMPROVING WEIGHT AND SPACE REQUIREMENTS.

*High Speeds.* Further means that may be employed to improve the weight and space requirements do not differ materially from those used on four-cycle engines and already reviewed. High rotative speeds as in the Attendu and Peugeot engines and a number of two-cycle submarine engines built by Sulzer Bros., Fiat Co., and by two French builders, have been employed. Unfortunately there is no published information available, although several different engines up to 6000-7000 horsepower[81] have been reported in construction for the French Navy. The 2500-horsepower Busch-Sulzer engine[82] installed in three U. S. fleet submarines is an example of this class. The Knudsen-Baldwin engine, which has already been discussed, makes also efficient use of the high rotative speed for weight reduction.

*The Double-Acting Principle.* The use of the double-acting principle in connection with two-cycle engines is spreading rapidly, and in the minds of a large number of oil-engine builders is the only means for meeting the

[80] "Engineering, Aug. 25, 1922, p. 226.
[81] "Motorship" (N. Y.), Feb., 1926, p. 121.
[82] JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Aug., 1925, p. 583.

demand for high-capacity engines. Vickers Ltd., for instance, after building a 2900-horsepower single-acting four-cycle engine of their own design, purchased a license for building M.A.N. two-cycle double-acting engines. The Werkspoor Co. is reported to be experimenting with a double-acting, two-cycle engine at their home plant, notwithstanding the fact that they collaborated with their British licensee in building double-acting four-cycle engines.

*Crank-Mechanism Modifications.* Modifications in the standard crank mechanism play also an important role in the development of the two-cycle engine, such, for instance, as Junkers and Fullagar adopted. In this connection mention should be made of the Michel engine that is being developed in Germany. The original engine of this type is reported to have been in operation at the builder's works for four years. From the schematic representation in Figure 29 it will be noted that three cylinders with a common combustion chamber are placed radially, one having exhaust ports while the other two admit scavenging air. At the ends of the piston rods are placed rollers running in a cam-shaped groove which is part of the flywheel. The whole is essentially a high-speed engine with a 3-to-1 reduction built into it. It is evident that if such an engine can be made to work so reliably that accessibility becomes a secondary matter, this solution is of far-reaching importance. But difficulties peculiar to such an unusual design no doubt developed, since no commercial installations of the engine have been reported.

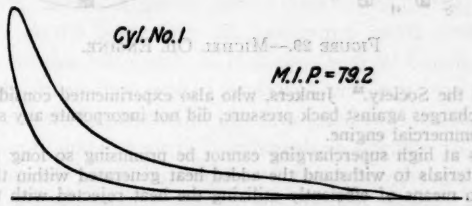


FIGURE 28.—INDICATOR CARD OF THE BUSCH-SULZER ENGINE, SHOWING SUPERCHARGING.

*Airless Injection of Fuel* has been used on two-cycle as well as on four-cycle engines. Attendu, Peugeot, and Knudsen have used it on high-speed engines, and Doxford, Scott-Still, and Junkers have applied it also and with very pronounced success, as already mentioned. A large number of rather small-capacity engines are using injection chambers, and the great majority of larger engine builders are still experimenting with airless injection, and the announcement of its adoption may be expected any time as evident from a recent paper by Dr. Eichelberg[53] of Sulzer Bros.

*Supercharging* is really a feature of the two-cycle engine and is employed in a large number of such engines. With valve scavenging this is obtained by leaving the valves open a short time after the piston passes the exhaust ports, and here it is extensively used, especially in altitude installations, where a slightly larger scavenging pump restores the engine to its normal output. In opposed-piston engines and in the Knudsen engine this can also easily be done by suitable timing of the two pistons. The Sulzer engine

[53] "Zeitschrift des Vereines deutscher Ingenieure," Aug. 7, 1926, p. 1079.

permits supercharging by virtue of the upper scavenging ports closing after the exhaust ports. Supercharging as applied by two cycle-engine builders refers to filling the cylinder with scavenging air up to a pressure of about 15–16 pounds absolute, and the effect of supercharging is clearly visible on the indicator diagram of the Busch-Sulzer engine, built for the Shipping Board[54] (Fig. 28). Attempts to increase supercharging led to heat troubles at the weakest spot in the engine, as told by Mr. Knudsen at the last Annual

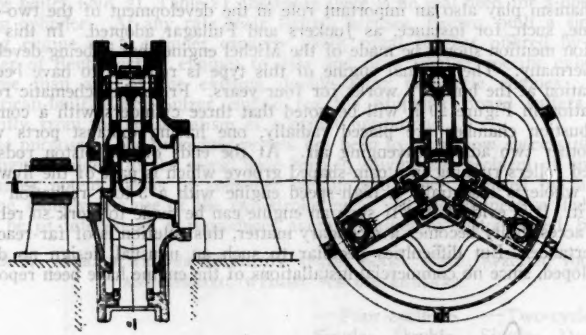

FIGURE 29.—MICHEL OIL ENGINE.

meeting of the Society.[55] Junkers, who also experimented considerably with high supercharges against back pressure, did not incorporate any such feature into any commercial engine.

Attempts at high supercharging cannot be promising so long as we have (*a*) no materials to withstand the added heat generated within the cylinder, and (*b*) no means of efficiently utilizing the heat rejected with the exhaust gases. These two considerations apply equally to four- and two-cycle engines, except that in two-cycle engines their need is greater. Thus the development of better heat-resisting materials will open up new avenues of experiment, and oil-engine development will follow.

*Compound Engine.* As to the utilization of the waste heat of the engine, aside from the rather inefficient use made of the exhaust gases in exhaust-gas boilers for heating purposes such as process water or building heating, this is done in the Still engine, where steam of 100 pounds pressure is generated from the jacket water by the exhaust gases and used on the under side of the piston, where by expansion it further abstracts heat and cools the piston. Except for the possible complication of the boiler, condenser, and feed pumps, this solution of the problem gives improved fuel consumption, though weight and space advantages are rather doubtful. Besides it appears to stand in real danger from the steam-engine advocates, who are apt to change it into a good steam engine with the oil engine only as a sort of standby.

When better materials are available the compound engine may become an attractive solution of the weight and space problem with good fuel consumption as an additional advantage. Compounding is nothing more than

[54] "Marine Engineering," Oct., 1926, p. 579.
[55] "Mechanical Engineering," May, 1926, p. 474.

supercharging of the high-pressure cylinder coupled with an efficient utilization of its exhaust in the low-pressure cylinder. So far, however, the efforts of Mr. Sperry have not reached commercial success, chiefly due to the failure of the transfer valve. And when restricted by the capacity of this part to withstand heat, the compound engine offers no advantages to compensate, for instance, for such a practical disadvantage as multiplicity of different parts.

CONCLUSIONS.

The foregoing study of modern oil engines can properly be summed up as follows:

*a* Substantial progress in oil-engine development is not made by unduly stressing one particular line of attack.

*b* The combination of a number of means tending to improve the horsepower output of a given engine structure is necessary.

*c* High piston speeds combined with high rotative speeds where practicable, airless injection of fuel, and the use of high-speed blowers for furnishing combustion air in both four-cycle and two-cycle engines are more commonly employed.

*d* The double-acting two-cycle engine, due to the simplicity of port scavenging, is regarded as being better able to solve the large-engine problem than any other type.—" Mechanical Engineering," Nov. 1926.

## BOOK REVIEW.

### WHAT ARE OUR FOREIGN POLICIES?

FOREIGN POLICIES OF THE UNITED STATES, THEIR BASES AND DEVELOPMENT. By JAMES Q. DEALEY OF BROWN UNIVERSITY. PUBLISHED BY GINN AND COMPANY, 1926.

At a time such as this when the United States is envied and disliked in Europe, distrusted by certain Latin-American nations and, at best, not understood in China, it is peculiarly fitting that introspective study be made of our policies toward those nations and interests with which we are in contact.

More particularly is such a study fitting to Naval Officers whose duties carry them throughout the globe and whose actions and attitude should be true reflectors of American policies and ideals.

There are very few American policies sufficiently definite that their application by our naval and diplomatic officials can be made a matter of political mathematics. Truly to apply American policies and correctly to exemplify American ideals, demands that the historical background of our domestic development be studied along with the growth of our international relations and the expression of the latter in treaties and understandings.

Professor Dealey traces this growth of America into a world power in interesting style and admirable brevity, leaving with the reader a very clear word picture of what "has been." He then sketches—but without the usual objectionable propheticalness—what he terms our "Changing policies" toward European, Latin-American and Asiatic affairs. Not the least of the very excellent material in this volume is the selected bibliography of recent works bearing on the United States and its world contacts which form in themselves a valuable compendium for more detailed study and research.

Many readers may feel that Professor Dealey is too optimistic as to our future relations in the Pacific and with Latin America.

This sentiment, particularly in regard to Pacific affairs, is shared by the reviewer. But there is at least one conclusion which can be drawn from the discussion and that is that "without an *adequate mobile modern* Navy the foreign policies of the United States are worth little if anything." The day of terrestial peace is still far away.

R. D. TISDALE.

# ASSOCIATION NOTES.

## ELECTION OF OFFICERS.

At a meeting of the Council of the Society held on December 29, 1926, votes cast for officers for 1927 were counted and the following were declared elected:

*President:*

Rear Admiral John Halligan, Jr., U. S. Navy.

*Secretary-Treasurer:*

Commander H. B. Hird, U. S. Navy.

*Council:*

Rear Admiral C. W. Dyson, U. S. Navy (retired).
Captain Ernest L. Bennett, U. S. Navy.
Captain Q. B. Newman, U. S. Coast Guard.
Captain S. M. Robinson, U. S. Navy.
Lieutenant Commander E. L. Cochrane (CC), U. S. Navy.
Mr. J. F. Metten.
Mr. H. M. Southgate.

## PRIZE ESSAY.

Stephen Elmer Slocum, a Civil Member of the Society, was awarded the prize for the best essay submitted in competition during 1926. The article, entitled "Practical Application of Modern Hydrodynamics to Marine Propulsion," is the first paper in this number of the JOURNAL. Mr. Slocum was born in Glenville, N. Y., in 1875. He received his B.E. degree at Union College, Schenectady, N. Y., in 1897, and the degree of Ph.D. at Clark University, Worcester, Mass., in 1900. He is a Fellow and Scholar of the latter university. He has held chairs at the University of Cincinnati, and the University of Illinois. He is now a Consulting Engineer with headquarters at Ardmore, Pa. Mr. Slocum has written a large number of books, research monographs and technical articles.

The prize will be offered again this year. It carries with it a gold medal, life membership, One Hundred dollars cash, and payment at current rates for the paper. The competition is open to all members, Naval, Civil, Associate and Life and it is hoped that many of our members will submit papers in this competition.

## FINANCIAL STATEMENTS.

Following are the financial statements for the year 1926:

### STATEMENT OF INCOME AND EXPENDITURES.

*Income.*

| | | |
|---|---|---|
| Publication: | | |
| Dues | $5,915.00 | |
| Subscriptions | 2,877.40 | |
| Sales | 1,137.42 | |
| Advertisements | 4,227.00 | |
| Exchange | .24 | |
| | | $14,157.06 |
| Interest on investments | | 889.49 |
| | | $15,046.55 |

*Expenditures.*

| | | |
|---|---|---|
| Publication: | | |
| Printing | $8,853.57 | |
| Engraving | 1,099.00 | |
| Drafting | 32.00 | |
| Manuscript | 823.00 | |
| Commission and discounts | 266.28 | |
| Postage | 163.03 | |
| | | $11,237.28 |
| Salaries: | | |
| Secretary-Treasurer | $1,500.00 | |
| Clerk | 1,100.00 | |
| | | 2,600.00 |
| Banquet, 1926 | | 217.78 |
| General expense | | 175.33 |
| Membership campaign 1926 | | 24.52 |
| Current profit and loss | | 253.00 |
| | | 14,507.91 |
| Net gain | | $538.64 |

BALANCE STATEMENTS.

| | | | |
|---|---|---|---|
| Assets December 31, 1926: | | | |
| Cash in bank | | | $2,690.06 |
| Accounts receivable: | | | |
| Advertisements | $1,843.77 | | |
| Dues | 551.25 | | |
| Sales | 696.92 | | |
| Subscriptions | 34.70 | | |
| | | 3,126.64 | |
| Notes receivable (N. W. Akimoff) | | 144.20 | |
| Investments (All Bonds): | | | |
| Armour Co. of Delaware | $887.50 | | |
| Army and Navy Club | 1,000.00 | | |
| Bethlehem Steel Company | 957.50 | | |
| Canadian Northern Railway Company | 1,115.00 | | |
| Hocking Valley Railway | 1,682.50 | | |
| International Mercantile Marine | 795.00 | | |
| New York Steam Corporation | 927.50 | | |
| Ohio Power Company | 860.00 | | |
| Penna. Power & Light Company | 882.50 | | |
| Swift and Company | 917.50 | | |
| Washington Ry. & Elec. Company | 4,230.00 | | |
| | | 14,255.00 | |
| Furniture | | 72.00 | |
| | | | $20,287.90 |
| Liabilities: | | | |
| Dues paid in advance | $142.90 | | |
| Subscriptions paid in advance | 937.74 | | |
| Advertisement paid in advance | 50.00 | | |
| | | | 1,130.64 |
| | | | $19,157.26 |
| Assets January 1, 1926 | | | 18,618.62 |
| Net gain | | | $538.64 |

A committee consisting of Captain Carlos Bean, U. S. N., Commander J. M. Irish, U. S. N., and Lieutenant Commander J. Douglas Jones, U. S. N., has been appointed to audit these accounts.

MEMBERSHIP.

The following members have joined the Society since the publication of the last previous JOURNAL:

NAVAL.

Custer, Garry D., Lieutenant, U. S. Navy.
Swain, Charles D., Lieutenant Commander, U. S. Navy.

CIVIL.

Stacey, Alfred E., Essex Fells, N. J.

ASSOCIATE.

Bernatto, John, Route 2, Box 40, Wiscasset, Maine.

ADDRESSES.

The Secretary-Treasurer is very anxious that you shall receive your copy of every number of the JOURNAL promptly. It is most earnestly requested that each member will give immediate notice of any change of address.